POWER

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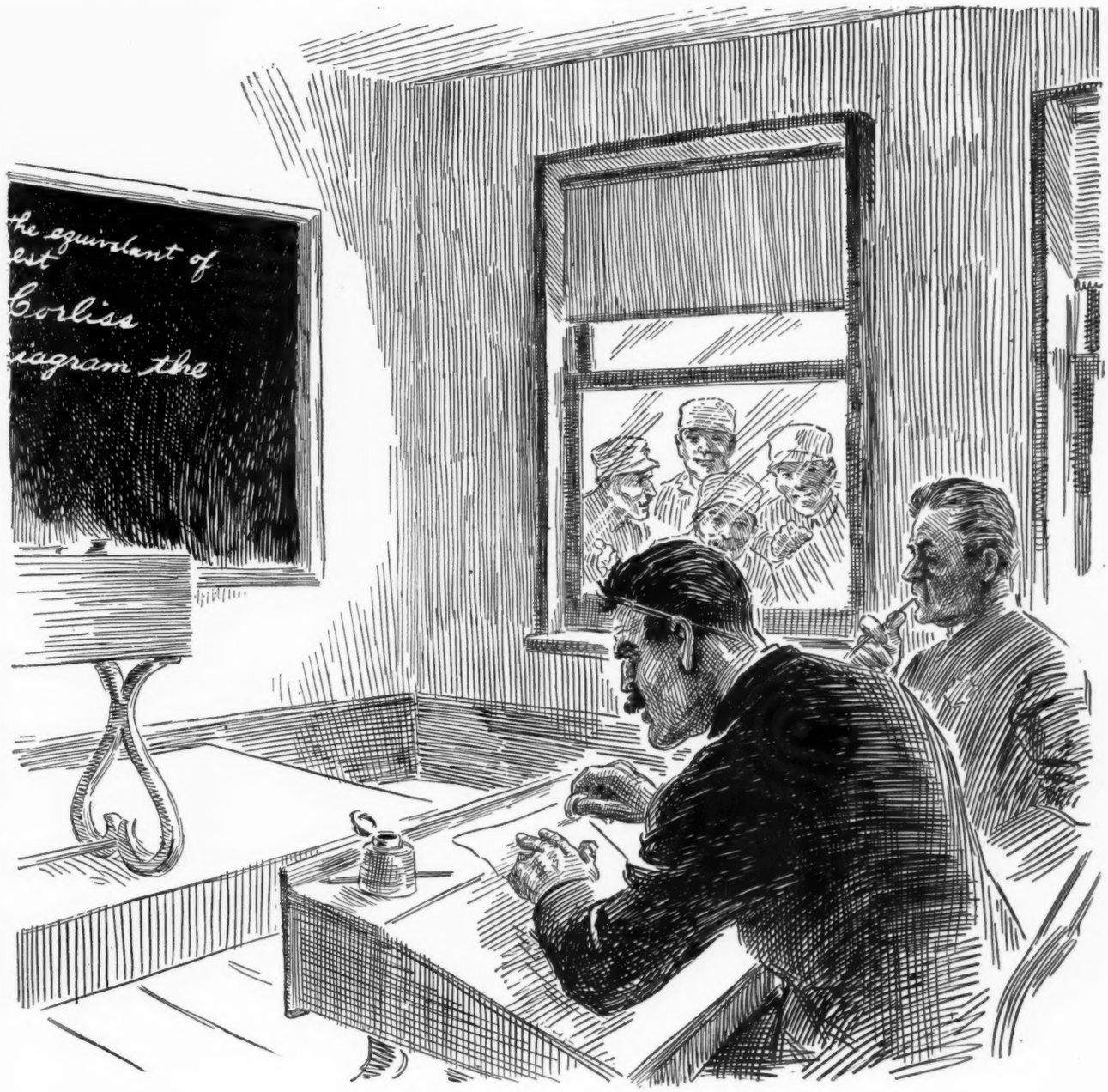
The Examiners Examined



THE TABLES TURNED IN MASSACHUSETTS

(*See pages 292 and 305*)

Turbines and Diesel Engines in Central Stations

SYNOPSIS—Abstract of a paper recently read by M. Gercke, chief engineer of the Maschinenbaufabrik Augsburg-Nürnberg, at a meeting of the association of electric central-station managers and engineers in Kiel, Germany. The author gives an interesting comparison of the relative economic importance of steam turbines and Diesel oil engines for central-station work.

In order to get a safe basis of comparison it is essential, besides considering the technical advantages and disadvantages of the two modes of power generation, to study their respective costs of operation.

Steam-driven Generators. Their chief domain is in

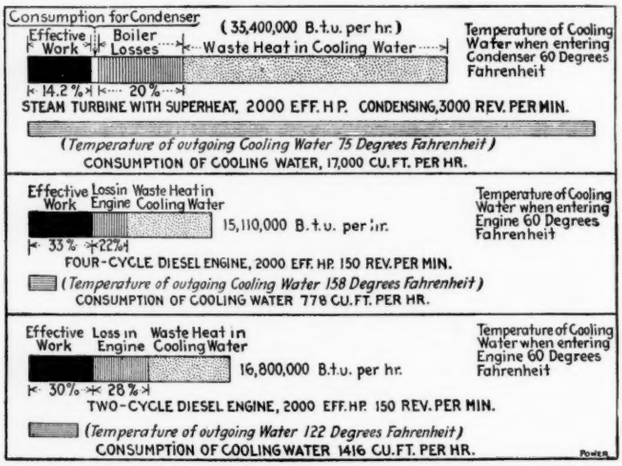

Consumption for Condenser
(35,400,000 B.t.u. per hr.)
Effective Work
Boiler Losses
Waste Heat in Cooling Water
Temperature of Cooling Water when entering Condenser 60 Degrees Fahrenheit
14.2 %
20 %
STEAM TURBINE WITH SUPERHEAT, 2000 EFF. H P. CONDENSING, 3000 REV. PER MIN.
(Temperature of outgoing Cooling Water 75 Degrees Fahrenheit)
CONSUMPTION OF COOLING WATER, 17,000 CU. FT. PER HR.
Effective Work
Loss in Engine
Waste Heat in Cooling Water
15,110,000 B.t.u. per hr.
Temperature of Cooling Water when entering Engine 60 Degrees Fahrenheit
33 %
22 %
FOUR-CYCLE DIESEL ENGINE, 2000 EFF. HP. 150 REV. PER MIN.
(Temperature of outgoing Cooling Water 158 Degrees Fahrenheit)
CONSUMPTION OF COOLING WATER 778 CU. FT. PER HR.
Effective Work
Loss in Engine
Waste Heat in Cooling Water
16,800,000 B.t.u. per hr.
Temperature of Cooling Water when entering Engine 60 Degrees Fahrenheit
30 %
28 %
TWO-CYCLE DIESEL ENGINE, 2000 EFF. HP. 150 REV. PER MIN.
(Temperature of outgoing Water 122 Degrees Fahrenheit)
CONSUMPTION OF COOLING WATER 1416 CU. FT. PER HR.
POWER

FIG. 1. HEAT UTILIZATION AND CONSUMPTION OF COOLING WATER OF STEAM TURBINES AND DIESEL ENGINES AT FULL LOAD

large power stations. Their advantages in general—and when turbines are employed in particular—are: unlimited capacity of units, low initial cost, as referred to the unit of output, large overload capacity, simplicity of attendance and upkeep, possibility of using all kinds of coal, even the most inferior grades, as, for example, raw brown coal, lignite, peat, etc. Their disadvantages are: dependence of heat consumption per unit output upon the character of attendance and upkeep, dangers and drawbacks associated with the use of boilers (cleaning, inspection, removal of ashes, dust and smoke nuisance), losses of fuel through starting and extinguishing fires, large consumption of cooling water by condensers, the operating condition of the latter being determinative for the economy of operation.

Oil-driven Generators. Their chief domain is in small- and medium-sized power stations. Their advantages are: high thermal efficiency, a low heat consumption per unit of output, which is practically independent of size of engine and skill of attendance; absence of smoke and fumes, immediate readiness to start, no fuel losses when starting or stopping, no trouble with dust in flues and removal of ashes; reduced purchase, replenishment and storage of fuel in tanks, less consumption of cooling water. Their disadvantages are almost nil, except the stresses exercised on the foundation by the inertia forces, which is a defect common to most reciprocating piston engines. Another drawback: Diesel engines cannot be built up to units of unlimited capacity. With horizontal twin tandem combinations the upper limit of size is 4000 hp. With vertical engines an increase of capacity up to 6000 hp. is possible, but a still further increase of the size of units is at present out of the question, as the heaviest pieces of a 6000-hp. twin-tandem engine running at 94 r.p.m., weigh already 80 tons, and must be transported by special railway cars.

Total Efficiencies. The above mentioned thermal superiority of the Diesel oil engine is not in all cases associated with a general economic superiority of the plant. The relative utilization of heat and consumption of cooling water in Diesel engines and turbines of the same size are graphically compared in Fig. 1. The figures refer to engines of about 2000 hp. output, latest design and full load. But in order to arrive at just conclusions as to the actual conditions of heat utilization, the heat consumption of the two types at partial loads must also be considered. Figs. 2 and 3 show that with partial loads the factor of fuel cost becomes less favorable in Diesel

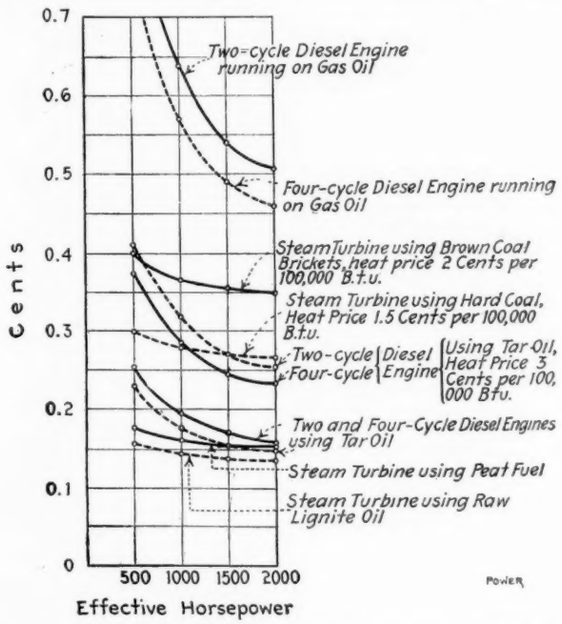

0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
Cents
Two-cycle Diesel Engine running on Gas Oil
Four-cycle Diesel Engine running on Gas Oil
Steam Turbine using Brown Coal Brickets, heat price 2 Cents per 100,000 B.t.u.
Steam Turbine using Hard Coal, Heat Price 1.5 Cents per 100,000 B.t.u.
Two-cycle Diesel Engine
Four-cycle Diesel Engine
Using Tar Oil, Heat Price 3 Cents per 100,000 B.t.u.
Two and Four-Cycle Diesel Engines using Tar Oil
Steam Turbine using Peat Fuel
Steam Turbine using Raw Lignite Oil
500 1000 1500 2000
Effective Horsepower
POWER

FIG. 2. HEAT PRICE IN STEAM TURBINES AND DIESEL ENGINES PER EFFECTIVE HORSEPOWER AT FULL AND PART LOADS, DISREGARDING THE EFFICIENCY OF THE ELECTRIC GENERATOR

plants. It should be added, however, that the operating conditions of the steam plant are assumed to be very favorable, namely, admission pressure of steam 12 atmospheres (176 lb.), and temperature 325 deg. C., with normal delivery of cooling water. If conditions are less favorable, the consumption of steam increases considerably.

Total Operating and Fuel Cost. The influence of fuel on the total operating cost depends on the following items:

(1) Heat consumption per unit of output.
(2) Heat price.
(3) Engine capacity.
(4) Ratio of actual operating hours to the maximum

possible operating hours per annum. In other words, the influence of fuel cost is proportional (1) to the local fuel cost, and (2) to the load factor

$$\frac{\textit{Rated capacity} \times \textit{annual operating hours}}{24 \times 365}$$

The higher these values are the greater is the importance of the fuel cost in the total operating expenses. The other items are: interest and amortization of initial capital, administration and attendance, lubrication and cleaning materials, upkeep and repair. The influence of fuel cost dominates with high heat prices and high load factor. The influence of interest and amortization dominates with low load factor and low heat prices, especially when plants serve as auxiliary or reserve. Therefore, Diesel engines are not adapted to the latter purposes, also

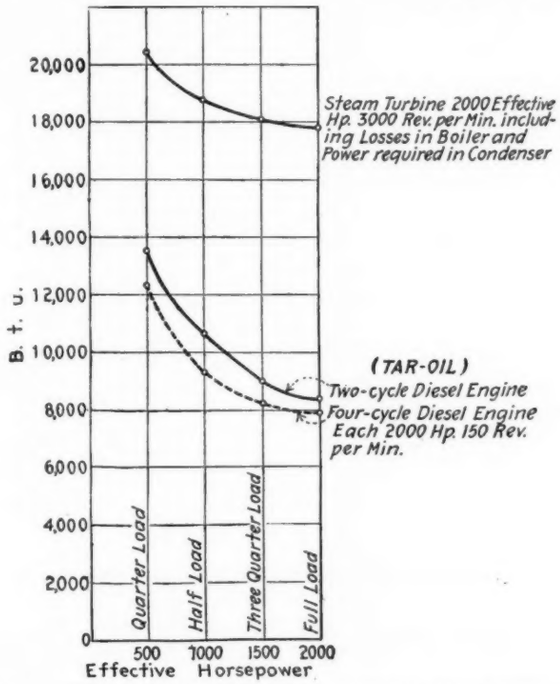


FIG. 3. HEAT CONSUMPTION OF STEAM TURBINES AND DIESEL ENGINES PER EFFECTIVE HORSEPOWER AT FULL AND PART LOADS, DISREGARDING THE EFFICIENCY OF THE ELECTRIC GENERATOR

not for covering occasional peak loads, because in this case the initial cost of the additional plant must be kept as low as possible, in order that the high interest and amortization charges may not burden the few operating hours of the auxiliary. An exception to this rule obtains in cases where immediate starting ability is required. Then the Diesel oil engine is superior to the steam turbine.

There are some cases which require special consideration, for example, when a central station is located directly at the mouth of the coal pit, which is a growing practice nowadays, or when it is situated in the midst of lignite and peat bogs, or in the direct vicinity of oil wells. In this case the fuel cost is very low and the steam turbine is superior to every other power generator, because its initial cost is small. The same holds true in the case of an electric power station which is to utilize the waste gases of blast furnaces, provided there are no steel plants or rolling mills attached to the furnaces which can use the surplus energy. Even though a gas-engine plant consumes only half the quantity of gas that the turbine-driven unit does for the same output, the latter is superior, all points considered. For this very reason there are more steam turbines and engines used in the Baku and Galician oil fields than Diesel engines.

Another case where the steam turbine is superior to the Diesel engine is when the waste heat of the plant can be utilized, especially when steam is used in considerable quantities for heating purposes. In central-station work this will hardly be the case, though in some cities, Nürnberg, for example, the waste steam of the central station is used in a public bath. If all the waste steam can be utilized the turbine becomes equal in thermal efficiency to the Diesel engine and superior in economic excellence, on account of the lower initial cost and the cheaper absolute fuel price. In the case of small Diesel plants the utilization of waste heat does not pay, because the amount available is too small to produce a sufficient quantity of steam or hot water (about 300 to 400 calories per effective

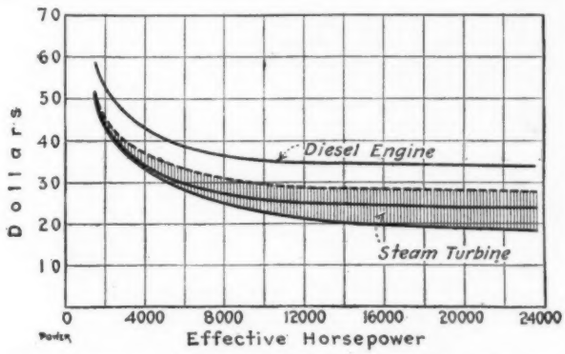


FIG. 4. INITIAL COST PER EFFECTIVE HORSEPOWER OF COMPLETE POWER STATIONS, NOT INCLUDING REAL ESTATE, IN GERMANY

horsepower-hour). In the case of larger Diesel plants the hot cooling water is preferably used in circulating heaters, and the heat of the burnt gases is utilized in economizers for preheating the feed water of boilers, as is done in the central station of Halle a.S. Finally, the Diesel oil engine is not suitable when units of over 4000 kw. or 6000 hp. are to be installed. In all other cases a comparative computation must ascertain whether a pure turbine or a pure Diesel plant is to be used, or whether a combination of the two types gives the highest economic efficiency. This is particularly necessary when an existing steam-driven central station is to be enlarged and accurate operating results are available to form the basis of comparative calculations.

Initial Cost of Complete Stations. In addition to the above reproduced curves showing heat price, heat utilization and consumption of cooling water, Fig. 4 gives a graphic idea of the comparative initial cost of the two types per effective horsepower for all sizes. The prices given refer, of course, to German conditions, but are nevertheless interesting. The number of types of Diesel engines among which a selection may be made being comparatively small, the curve of initial cost is simple. In the case of the steam turbine a great deal depends upon the selection of the units; hence the initial costs are represented in form of a strip, the lower limiting curve of which gives the minimum values, and the middle curve the normal values which may be attained. The cost of real estate and the initial expense for water supply or

cooling-tower plants are not included in either case. If these are required they are, of course, more expensive in turbine plants than in Diesel plants. The cost of attendance of both types may be considered as nearly equal, because the higher expenditures for waste and lubrication of the Diesel engine are counterbalanced to some extent by the additional expenditures for the attendance of the boiler house. In the case of large units, steam turbines are again greatly superior, in this relation, to Diesel engines. The total annual operating costs of electric central stations of 15,000 hp. = 10,000 kw., equipped with steam turbines and Diesel oil engines respectively, are graphically shown in Fig. 5.

Summary of Results. If a new central station of small capacity, say up to 1000 hp. total output, is to be built, the Diesel engine under normal conditions is the most advantageous mode of generation, unless special fuel conditions prompt the application of piston steam engines. Steam turbines are in no case advisable for small capacities. In the case of central stations of medium size, from 1000 to 6000 hp. total output, the steam turbine will prove superior in many cases. For capacities beyond 6000 hp. or 4000 kw. it is the only type to be adopted. Nevertheless, the large Diesel oil engine may prove an exceedingly valuable auxiliary to the steam turbine in medium and large central stations. Besides being economically comparable with the steam turbine in case of high load factor and high local fuel cost, even for large work, there is the immediate readiness to start and the avoidance of fuel losses during stoppages, which commend the Diesel engine for that purpose. Even the largest units can be started within two minutes and connected in parallel on the line, and loaded up to their full capacity. This is very convenient in case of a breakdown or a sudden increase of load in the central station. Small floor space, no special permit required, simple fuel supply, possibility of storing large quantities of fuel in tanks without loss, are other advantages of the Diesel auxiliary. If strikes or other interruptions in the supply of coal occur, the Diesel engine is extremely useful, because it may be operated independently, the oil being pumped in automatically, and no special attendance being required. The central station of Halle is remarkable and well worth visiting for the automatism of its fuel supply. The storage and supply of coal to the boilers, even if automatic, is much more expensive and largely dependent upon the attendants, who may impair the continuity of the operation. This danger is specially imminent if the bunkers in the boiler room are not very large. Moreover, the storage of coal involves considerable heat losses and liability to self-ignition and fire.

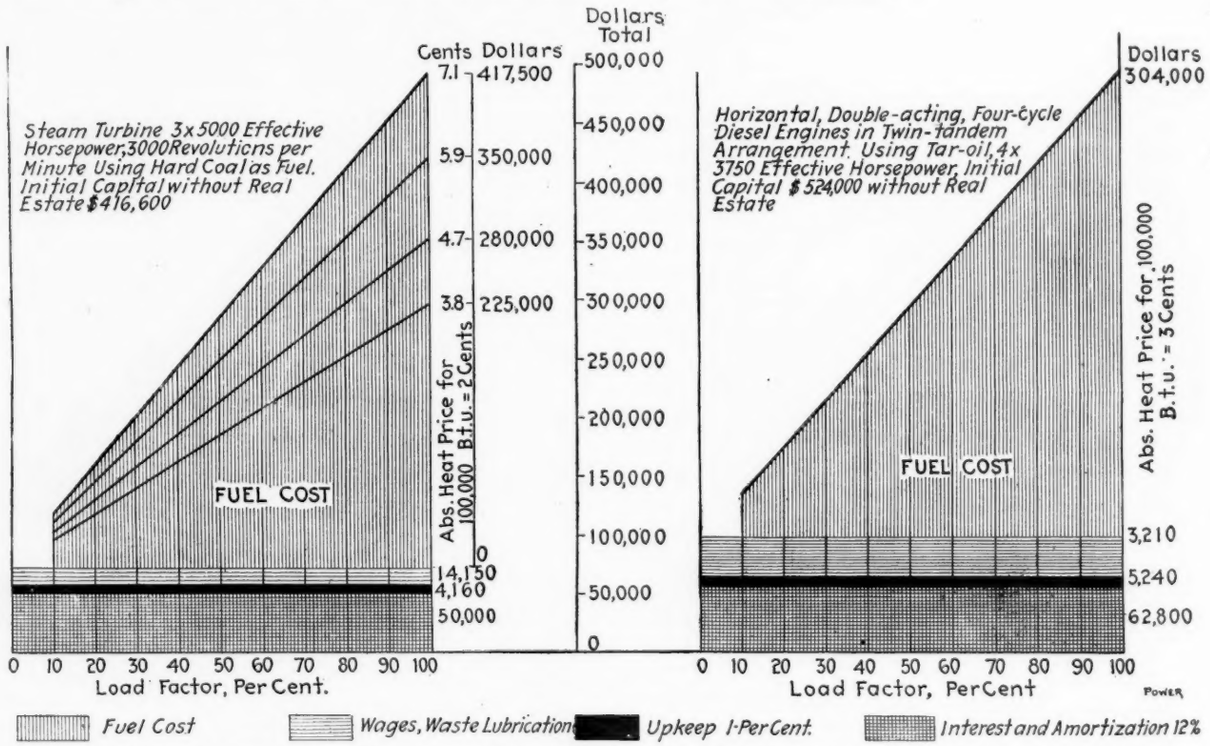


FIG. 5. ANNUAL OPERATING COST OF AN ELECTRIC CENTRAL STATION OF 15,000 EFF. HP. (10,000 KW.) TOTAL CAPACITY, IN GERMANY

The following conclusions are formulated by the author:

(1) For capacities of over 1000 hp. the initial capital outlay for a Diesel plant is higher than that required for a turbine plant of the same size. Hence the amount of interest and amortization per unit of output is also higher in the Diesel plant. These items, however, are of decisive importance only if the load factor of the station is low. For higher loads they drop out of the calculation. As far as the item of operating cost is concerned the Diesel engine is not to be recommended as an auxiliary. Only when the immediate readiness to start is a chief consideration is the Diesel engine superior.

(2) The cost of heat per unit capacity in Diesel engines is considerably lower than in steam turbines, provided the local price per 100,000 calories is 5c. or more, or if unfavorable water conditions prevent the utilization of the highest possible vacuum in the condenser of the turbine plant.

(3) With low load factors and low local fuel cost a steam turbine is the only mode of generation to be employed for large work, never a Diesel engine.

(4) With high load factor and high local fuel cost the case is reversed, a Diesel plant being more economical.

(5) The advantages of both types may in many cases be combined by installing Diesel engines and steam turbines together, the former taking care of the continuous load or part of it, and the latter of the peak loads, occurring during few hours of the day. This distribution commends itself on account of the lower initial cost of the steam turbine and of the possibility of augmenting the output of high-pressure boilers for some time. Quite a number of combined plants of this character are already in operation on the European continent.

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Smoke Prevention

By A. C. Wilson

SYNOPSIS—Some suggestions for burning coal with a minimum of smoke. The author points out that the chief difficulty as regards smoke legislation is to prove that a chimney so habitually emits smoke as to be a nuisance.

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Increased attention is being paid every year to the public nuisance caused by smoke from factory chimneys, but considerable ignorance as to the real problem involved is often shown by those most active in complaining of the smoke nuisance. Such legislation as exists only imperfectly provides for the detection of offenders.

Among other fallacies are the following: (1) That smoke is caused by incomplete combustion of coal due to insufficient air being supplied to the furnace; (2) that the loss of heat due to incomplete combustion represents a considerable proportion of the total heat in the coal and (3) that smoke can be easily prevented.

As regards legislation, the principal difficulty is to establish a basis on which it can be easily proved that a particular chimney so habitually emits smoke as to constitute a nuisance. Such laws as exist either leave each case to be fought out on the general ground of whether a nuisance exists, or, if it is attempted to make any clearer definition, the emission of black smoke is prohibited.

Even if there could be no question as to whether smoke is black, smoke of less density may be as great a nuisance as black smoke in respect to pollution of the atmosphere. The nuisance is caused by sending particles of unburnt carbon or hydrocarbons into the air, and the density of smoke may be reduced to almost any degree by simply diluting it with air without altering the total amount of solid matter emitted. To compare the amount of preventable smoke discharged from two chimneys it is necessary to know the amount of coal being burnt in the furnaces. Many so called smoke-preventing devices simply aim at diluting the gases with excess air during the time smoke is most likely to be given off.

Very few boiler furnaces fail to supply sufficient air for the complete combustion of the coal fired in them. About 12 lb. of air is necessary to supply the oxygen for completely burning 1 lb. of coal and under actual conditions most furnaces pass from 20 to 30 lb. of air per pound of coal. The reason smoke is emitted is that the air is not intimately mixed with the carbon and is not at a sufficiently high temperature.

The problem of insuring complete combustion of bituminous coal is mainly one of temperature and involves continually keeping the furnace at a high temperature, which is by no means easy. In the first place, the gases are reduced in temperature by coming in contact with the comparatively cold boiler plates; again, the furnace temperature is lowered every time fresh coal is thrown on the fire. One must, therefore, aim at preventing any gases from coming in contact with the heating surfaces except such as are products of complete combustion and also at introducing fresh coal in such a manner as to avoid reducing the furnace temperature either by reason of the low temperature of the coal itself or by the rush of cold air accompanying it.

To fire in such small quantities that the amount of green coal on the bars at any time is insufficient to appreciably reduce the furnace temperature would involve continuous work such as no human being would perform.

Possibly the best attempt at smokeless hand firing can be attained by adopting the coking system, that is, firing green coal only on the dead plate and the front part of the grate and gradually pushing it to the back after all the hydrocarbon gases have been driven off and it has become an incandescent mass. This method of firing, however, is impossible unless the boiler is worked at a low rate, as the coking process, during which the hydrocarbons are given off, is comparatively slow and the effect is to reduce the available grate area of the boiler by about one-half. If boilers are worked at a high rate it is impossible to fire them by hand without smoke and the only practicable cure generally is to increase the number of boilers.

Mechanical stokers more nearly comply with the conditions for smokeless combustion because the coal is fed without opening the fire doors and in small quantities at a time, if not continuously. That they do not always succeed in working without producing smoke is generally due to the attempt to force the fires. This results in a low furnace temperature because there is too much green coal on the bars.

A sharp draft is absolutely necessary for maintaining a high furnace temperature and the greater the amount of coal burned per square foot of grate, the sharper must be the draft, otherwise a fire is obtained which is incandescent underneath and more or less green on the top. Further, the draft must be well distributed or the fuel bed will develop holes, as happens with some types of forced-draft furnaces where the draft is produced by blowers or steam jets.

Such furnaces often give a fire having several bright patches in a bed of green coal and the half-burnt gases from the green coal never become intimately mixed with

the hotter products and consequently pass away as smoke. Unless smoke is consumed in the furnace it never will be consumed; it may be so diluted as to be light in color, but the objectionable matter is being expelled into the air just the same.

As to the common fallacy that smoke necessarily represents a large waste of fuel, a chimney may be smoking badly and yet the amount of heat lost by incomplete combustion may be negligible. The writer has tested the gases at the base of a number of smoky chimneys and has found that an average of 1 per cent. of carbon monoxide in the gases is very seldom reached. As high a percentage as this is only obtained when there is little dilution with excess air and when the carbon dioxide present is high, say, 10 to 12 per cent.

Assuming the composition of the flue gases to be carbon dioxide (CO_2) 12 per cent., carbon monoxide (CO) 1 per cent., oxygen 6 per cent., nitrogen 81 per cent. and the calorific value of the fuel to be 12,500 B.t.u. per lb. the heat lost by incomplete combustion represents about 5 per cent. of the total available heat in the fuel, which is much less than many imagine.

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Power Transmission by Compressed Air*

SYNOPSIS—Why compressed air has been considered as an inefficient means of power transmission and improvements made by manufacturers which have brought it to an equality with electrical transmission.

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In steam-driven machines of the oldest type the horizontal design was used, with the steam and air cylinders in tandem. No water jackets or spray cooling arrangement were provided for the air cylinder, and the piston speeds averaged from 200 to 300 ft. per min. The air valves were of the heavy flat type or of rubber, clearance spaces were anything and the compressor single-stage.

Present-day designs are mostly vertical and of the inclosed type employing forced lubrication and water jacketing for the cylinders. For pressures from 70 to 100 lb. the compressor is generally two-stage. In a general way the above description covers the major portion of compressed-air plants with piston speeds up to 500 or 600 ft. per min. Valve gears of combined mechanical and automatic type, or of light multiple-ported small-lift types, are employed and clearance spaces are down to 2 or 3 per cent. When in some instances old compressors of the first mentioned design, or somewhat similar, were discarded and a new plant installed, the results obtained easily explain why compressors have for long been considered inefficient. To demonstrate what can be done by installing new machines a case is mentioned where a compressor of four times the capacity of the old machine taken out required only the same amount of steam per hour.

When considering the general question of compressed air efficiency it is necessary not only to examine the compressor, but the pipe lines and the plant consuming the air must be considered; in this direction there has been great improvement. Taking a wide view of the matter few engineers have given much study to compressed air and have taken for granted that what has been must still continue. Looking at the matter from this standpoint, what is generally found in inquiries issued for large compressors? The capacity of the machine is given; if steam driven, steam pressure and vacuum are mentioned; if electrically driven, voltage. It is not general that steam consumption is asked, and on very rare occasions is a capacity test called for. Such a test when made is seldom, if ever, any check on the output of the machine, the test generally being to fill a reservoir of given capacity from atmospheric pressure to final working pressure. The temperature of the air may be taken or may not, and when taken is generally misleading and possibly allows of error up to 10 per cent. In fact it is questionable whether there is one compressor at work at the moment, the actual capacity of which is known correctly through a test.

The actual proof that any given machine is really delivering the full amount of air specified is not, of course, sufficient to prove the machine the best possible, as some designs may require more power to compress the air, due to small restricted air passages, poor cooling arrangements, low mechanical efficiency, etc. On the other hand a machine may not have a long life at a high state of efficiency. After all what the user wants is a machine to show small horsepower for air actually delivered, and a figure representing horsepower to compress say, every 100 cu.ft. of air, covers volumetric efficiency, mechanical efficiency, losses in compression, etc.

Taking a reciprocating two-stage air compressor, it will be found that a figure such as 20.5 electrical horsepower is required to compress and deliver every 100 cu.ft. of air up to 100-lb. pressure. Such a figure might reasonably be expected as "covers" for, say, 90 per cent. motor efficiency. For steam-driven machines the indicated horsepower in the steam cylinder per 100 cu.ft. of air compressed as above, should be about 20.

Passing to present designs and bearing in mind the present tendency to install more of such plants in collieries and shipyards, it might not be out of place to give an idea of the general tendencies. First it is found in a general way that colliery units range from about 1000 to 5000 cu.ft. and the electrically driven units from 1000 to 3000 cu.ft. with unit horsepowers from, say, 200 to 600. These represent plants placed on the bank and power drawn from local power companies' mains. In some quarters there seems to be a tendency to try in the mine smaller semi-portable sets of strong and efficient design, with more attention given to strength and simplicity than was hitherto the case, when lightness was considered the main necessity.

Comparing overall efficiency of the electrical methods with a large steam-driven compressor on the bank, Mr. Barr gives the following losses. In the steam-driven machine the mechanical loss is 10 per cent. and transmission losses including leakage 7 per cent. For the electrical plant assume the current generated at the power house, as it will be found in a general way that the losses in the generation will equal the extra charge for the power as made by the power company. Then there are 10 per cent.

*Abstract of paper read by George Barr before the Manchester (Eng.) Association of Engineers.

mechanical loss in the engine, 10 per cent. loss in the dynamo, 5 per cent. loss in the transmission for the small semi-portable machines and 10 per cent. loss in the motor driving the compressor. For large motor-driven machines the same allowance of 7 per cent. loss for air transmission must be made as in the steam-driven sets, and for the small semi-portable sets there is a loss of 3 per cent. for the short air piping and leakage. These figures are all relative, but are a fair mean for all the cases. Steam losses therefore equal 17 per cent., large motor sets 37 per cent., and small motor sets 38 per cent., or a difference of 20 and 21 per cent., respectively, representing a margin to cover extra price paid to power companies, or a figure which will make it possible to decide whether extra interest could not be readily paid on the extra capital for the steam-driven sets, together with the difference in running cost and upkeep of both schemes.

Mr. Barr reviewed the changes which have taken place in detail of construction, such as cooling arrangements, governing, lubrication, valve gear, etc. Summarizing he finds with a large installation of four machines of, say, 2500 cu.ft. capacity each, first, no automatic stopping gear is wanted, as there will be an attendant who can be made use of for any starting or stopping actually wanted over a given day. But if in an electrically driven plant one of the sets had a variable-speed motor to give a desired range when all the plant was in use, this would cover all actual requirements. However, an additional controlling gear arranged to allow so much air to be discharged before compression starts, would give a plant covering all possible fluctuations and assure high efficiency over varying conditions, with the extra advantage of being reasonably cheap.

At the moment automatic valve gears are those most used, and certainly it would appear that the greatest progress has been made in this class. They take the form of a multiple-ported or grid valve with very small lift, the form of valve, fixings and springs being covered more or less by various patents. Should an examination be made of indicator diagrams taken from one of the latest high-speed machines with multiple-ported valves, it would be apparent that the losses due to throttling either on the inlet or discharge are very small and leave little margin for improvement in this direction.

Where it was common to install single-stage machines for pressures of say, 60, 80 and even 100 lb., two-stage compression is now most common. An increase in compressor efficiency of say, 10 to 15 per cent. is obtained in large size units, which also makes for greater reliablity due to absence of heating troubles. Further, cooler and drier air in the system can be secured, but it is necessary to have intercoolers of reasonable size, 1 sq.ft. of cooling surface for every 4 cu.ft. of air compressed to 100 lb. not being too large. The intercoolers are also better if placed outside the engine rooms in the atmosphere, say on the shaded side of the building. The cylinder cooling in the best makes of machines may be taken as generally the most efficient possible, as most compressors now have jackets carried entirely around the cylinders and covers.

Referring to the question of reheating, Mr. Barr admits that it would certainly add considerably to the efficiency of a compressed-air installation. In collieries however, the air is not used at one particular spot; to heat the air immediately after leaving the compressor to, say, 350 deg. F. and then pass it along a distance of perhaps 1000 ft. is simply wasting money.

With newer designs of machines where piston speeds are higher, more care must be taken of air-cylinder lubrication. In fact, more trouble develops from the use of unsuitable oils in the cylinders than is necessary. The same conditions do not prevail in an air cylinder as in the average steam cylinder, where condensation can always be depended upon to assist towards lubrication. In the air cylinders, although there is moisture in the air, this will not deposit at the high temperature. In fact, the air cylinders will run very dry and seizures will be the rule if proper care is not taken. For air-cylinder lubrication an oil with high flash point is needed, which will leave little or no deposit, such as high-class gas-engine oil.

These remarks do not refer to compressors other than the usual machines for pneumatic tools and uses requiring similar pressures. For higher pressures machines of three- and four-stages are necessary.

Air compressors can now be had very much higher in efficiency than ten years ago. More care is being taken in the layout of piping systems, and if this betterment were carried into the plant which consumed the air, there would be shown in such places as collieries, results equal to the best electrical schemes. This statement does not suggest that over an instantaneous reading an overall efficiency could be shown with air equal to electricity, but taking a year's comparison with all upkeep charges, running cost and interest on differences in invested capital, or in other words the cost per ton of coal raised averaged over a year, the costs would be much the same. This is the correct test and the efficiency which really matters.

Power Plant Operation in the Army

Brig.-Gen. Weaver, chief of the Coast Artillery Corps U. S. A., in his recent report to the Secretary of War, has this to say of the corps method of training its men in the operation of the electric power plant:

The War Department some years back provided courses of instruction for officers and enlisted men at the Coast Artillery School at Fort Monroe, Va., that had for their object precisely the training of selected officers and enlisted men for these technical duties.

Accordingly there are among the commissioned personnel of the coast-artillery officers who are designated as "artillery engineers," and among the enlisted personnel noncommissioned staff officers with the grades of "engineer," "fireman," "master electrician," "electrician sergeant" (first and second class) and "master gunner," whose duties relate particularly to the care, preservation and maintenance of the electrical and mechanical equipment in condition of thorough serviceability.

Based on the estimates of the National Coast Defense Board, the percentages of completion of electrical installations necessary for full protection of home ports are: Search-lamps, 41 per cent.; power plants, 50 per cent., and submarine mines and structures, 81 per cent.

The power plant at Fort Monroe is operated entirely by enlisted men of the Coast Artillery Corps under the supervision of the head of this department of the school.

The results obtained in the operation of this plant have been very satisfactory, an average daily power and lighting output of 1200 kw.-hr. for the 12 months

ended Feb. 29, 1912, having been produced at a cost per kilowatt-hour at the switchboard of 3.38c., including all supplies, repairs, labor and depreciation, or of 1.19c., excluding labor and depreciation costs.

It is estimated that the installation of this plant and its present use for post lighting, pumping, fortification, school and heating purposes has resulted in a decrease in the total annual maintenance cost of the school and post of Fort Monroe of approximately $25,000; or, making allowance for labor of all enlisted personnel involved and for depreciation cost of plant, is an annual net commercial saving of over $13,000.

It is indicative of the high opinion held of this plant in commercial circles that some of the most prominent manufacturers of engineering apparatus in the United States have cited it as exemplifying in a noteworthy manner the efficiency that can be obtained with isolated plants of comparatively small capacity.

❖ ❖

The Examiners Examined

The position of chief of the boiler-inspection department of Massachusetts, which department has also charge of the examination and licensing of engineers, became vacant in July of last year and in October a civil-service examination open to members of the department was held. Only seven of the nineteen examiners took the examination and only three of these obtained the 65 per cent. necessary to qualify.

The following list of questions is sent to us as those asked upon this occasion. They do not seem to overweigh in difficulty those which the examiners are in the habit of asking, as evidenced by questions said to have been asked by them and sent to us for solution.

Civil Service Examination
held Oct. 14, 15, 1912, for
Acting Chief Inspector of Boilers.
Commonwealth of Massachusetts.

Mathematics: (Given weight of 3 in 15)

9 practical problems, including one to be solved by logarithms.

Spelling: (Given weight of 1 in 15)

20 words selected from the Boiler Inspection Law.

Letter: (Given weight of 2 in 15)

To consist of 300 words written on a choice of three subjects, namely—What are the prime requisites of a first-class engineer? Why did you select engineering as a life work? Which is the most important qualification in an engineer; mechanical ability or good morals?

Penmanship, including copying: (Given weight of 1 in 15)

Copy a section of law.

Practical and Technical Examination: (Given weight of 8 in 15)

Consisting of the following 36 questions:

1. Sketch all the essential parts of a compound-spring steam gage, registering both pressure and vacuum.

2. How many B.t.u. are required to change one pound of ice at zero F. to steam at atmospheric pressure?

3. What is meant by absolute zero of temperature, and how many degrees below zero on the Fahrenheit scale is it?

4. Show by figures what height a column of water will be balanced by a pressure of 14.7 lb.

5. The low-pressure cylinder of a cross-compound engine is doing more work than the high pressure. State fully how you would divide the work evenly between the two cylinders.

6. Do these indicator diagrams (representing one diagram taken from each end of the cylinder of a steam engine without taking the paper from the indicator drum) show that at any time during the stroke the resistance to the piston's forward motion is greater than the effort tending to move it forward? If so, indicate the point where this is so and show by figures approximately how much the resistance exceeds the effort and also state fully why the engine does not stop at or near the point you have indicated. Use pair of ordinary slow-speed four-valve cards.

7. The temperature of steam supplied to an engine is 331.169 F. and the temperature of the exhaust is 219.452 F. What is the thermal efficiency of the engine?

8. If a condenser is added to the above engine so that the temperature of the exhaust is lowered to 153.123 F., what will be the increase in thermal per cent.?

9. As the difference in temperature of the steam on entering and leaving the cylinder of a steam engine is increased, what source of waste is greatly increased, and how can this waste be partially avoided?

10. A vacuum gage on the condenser of a steam engine shows a reading of 27 in.; what is the absolute pressure in pounds per square inch in the condenser? Barometer reads 30 in.

11. What is the most accurate way of ascertaining the steam consumption of a steam engine per indicated horsepower, and how may the steam accounted for by the indicator be obtained from the indicator diagram? What is the value of such a determination?

12. Name the various places where an indicator may be advantageously attached and state what information you would expect to obtain from cards taken from the places you have named.

13. What does the term "slip" mean as applied to a pump? With a pump in good working order, about what per cent. of the capacity of the pump does the slip amount to?

14. What does the term "pump duty" mean? Give an example.

15. How would you determine how large a feed pump would be required to feed the boilers of a given plant?

16. In feeding a boiler with an injector against its own steam pressure, where does the power come from, and could the same injector be used to feed a second boiler carrying steam at a higher pressure?

17. How would you calculate the size of a steam main needed for a battery of boilers using 10,000 lb. of soft coal per hour. The highest pressure carried is 150 lb. absolute and the lowest 100 lb. absolute. The specific volume to be taken as 3.1 and 4.4 respectively.

18. An horizontal tubular boiler with 30.27 ft. of grate is under forced draft and 50 lb. coal per sq.ft. grate per hour is burned. The boiler works at 150 lb. absolute pressure. Determine size of required safety valve.

19. Show the best method of staying the lower part of the rear sheet in a cylindrical multitubular boiler, having a manhole in the front head.

20. A plain slide-valve engine has ¾-in. outside lap on the valve and ¼-in. exhaust lap. Suppose one-half to be taken off inside lap at each end. What changes will take place in the events of the stroke? Explain by cards.

21. What tests are necessary to determine the suitableness of steel plates of steam boilers? State in full as to the proper requirements, physical and chemical.

22. What is meant by priming in a boiler? Would the temperature of steam at a given pressure vary with different degrees of priming?

23. Steam from a 150-hp. engine using 26 lb. of steam per horsepower per hour exhausts into a surface condenser at a pressure of 4 lb. absolute, the temperature of the condenser water being 55 F. on entering the condenser and 95 F. on leaving the condenser. How many pounds of condensing water are required per hour for this engine?

24. A surface condenser is to be attached to an engine capable of developing (condensing) 350 hp. The amount of steam per hp. per hour is 19.5 lb. How much cooling surface should the condenser have?

25. What is the difference between the real and apparent cutoff of an engine and how is the real cutoff found when the apparent cutoff and clearance are known? Give an example.

26. The indicated horsepower of an engine is 563.74 and the net horsepower is 484.25. What is the mechanical efficiency of the engine?

27. Describe an inside and an outside fusible plug, and state why the distinction is made and where each should be used.

28. Why is it difficult to lift hot water on a long lift with an injector?

29. Why are two wristplates used on the low-pressure cylinder of a compound- or triple-expansion Corliss engine?

30. What do you understand by a "balanced valve" in a steam engine? Name some balanced valves.

31. When is steam said to be saturated and when superheated?

32. Design and calculate a double-riveted butt joint, furnishing your own data.

33. If a pop valve of a boiler is set at 100 lb. and the same fails to blow when the gage indicates 110 lb., what would be the first step to ascertain the reason for its failure to blow at 100 lb.?

34. Suppose two boilers of the same dimensions with equal grate area, one carrying 100 lb., the other 160 lb. Which requires the largest safety valve? Why?

35. State the possible advantages and disadvantages of the so called "high lift" spring safety valve as compared with the ordinary lift valve.

36. State in full how you would proceed with an investigation as to the cause of a boiler explosion, when there had been serious damage to property or loss of life.

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F. T. Monorail System

The F. T. (fixed tongue) monorail system consists of a single I-rail with the wheels of the trolley running on the lower flanges of the rail. The main feature is the means of switching from one track to another without the operator leaving the cab.

On approaching a track switch at which the operator desires to run from the main to the spur track, a steering lever on the trolley near the controller is pulled. This raises a horizontal roller to a position in which it engages

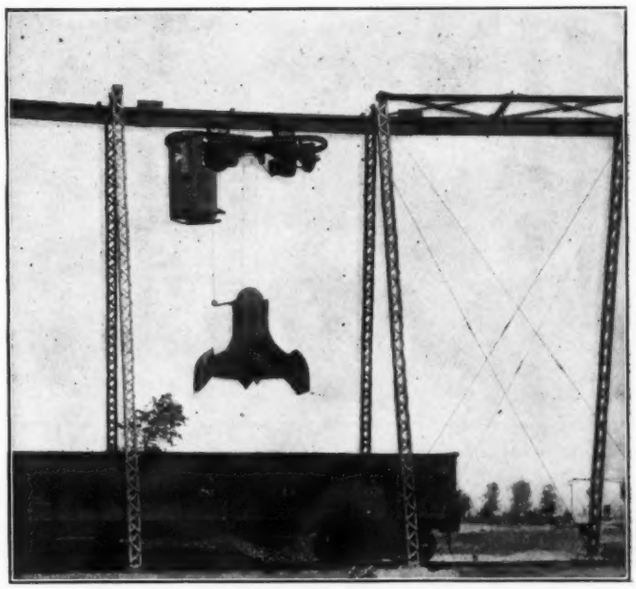


FIG. 1. UNLOADING A COAL CAR

a curved rib on the underside of the central switch tongue and swivels the leading track, thereby diverting the trolley onto the spur track.

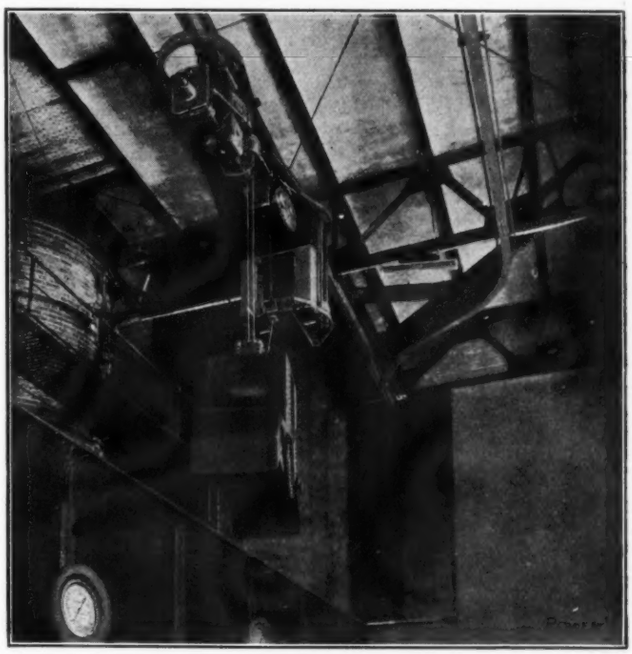
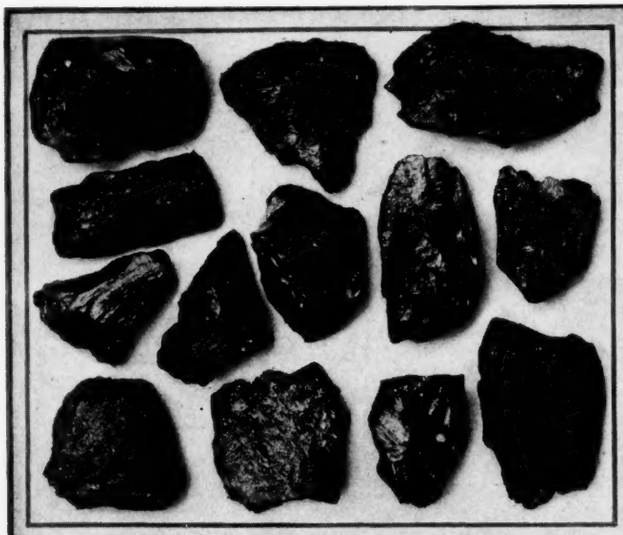


FIG. 2. DUMPING INTO THE STOKER HOPPER

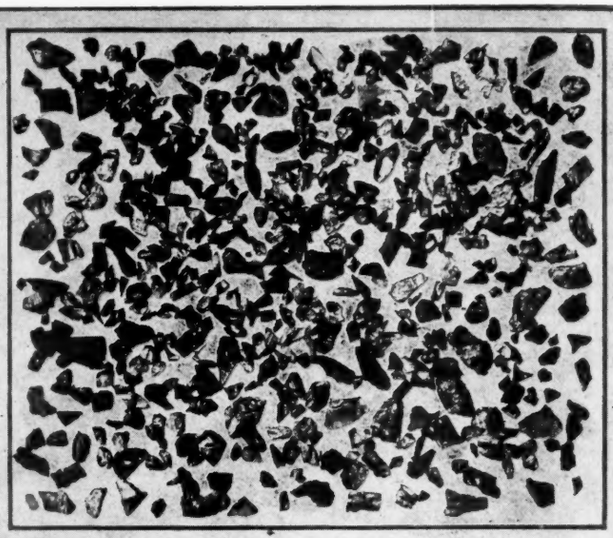
The system is suitable for power-plant work in unloading coal from cars, as shown in Fig. 1, and for dumping it into the stoker hoppers, as shown in Fig. 2, where a load has just been dumped.

This system is made by the Shaw Electric Crane Co., Muskegon, Mich., and handled by Manning, Maxwell & Moore Co., 85 to 89 Liberty St., New York City.

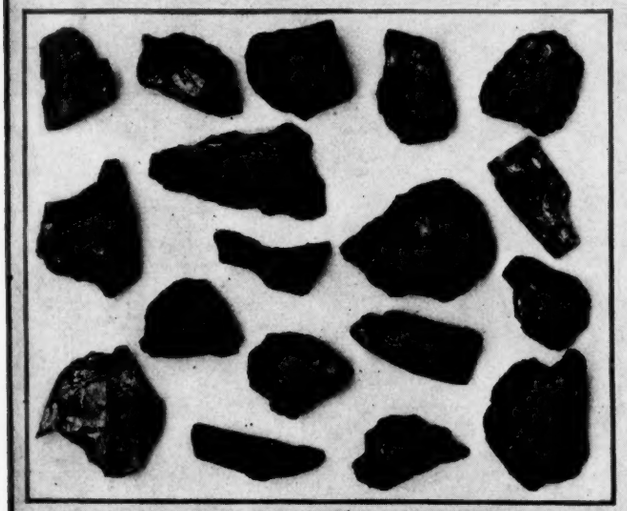
Anthracite Coal Sizes—Actual



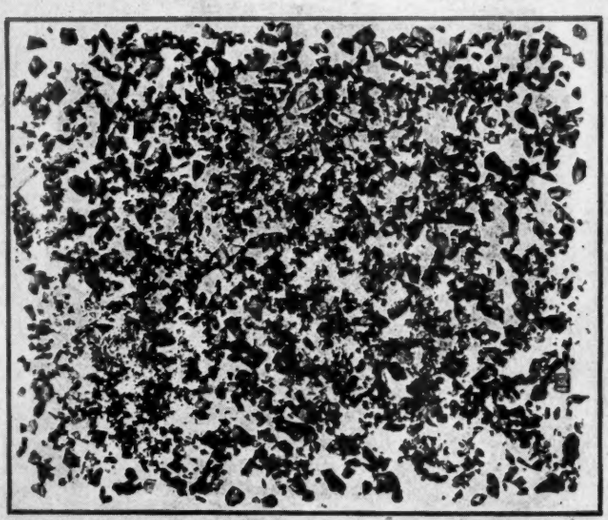
PEA



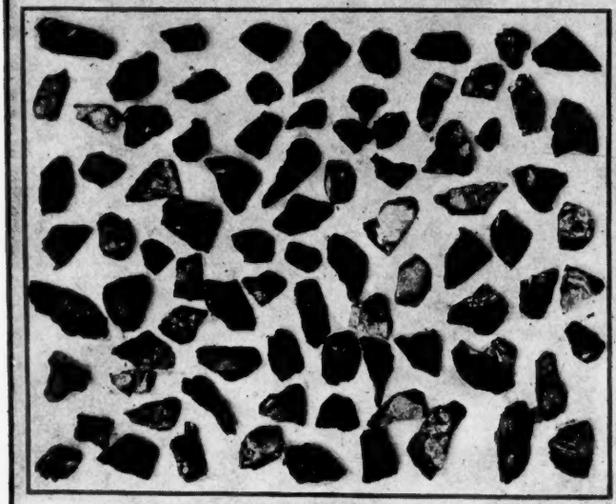
No 3 - BUCKWHEAT OR BARLEY



No 1 - BUCKWHEAT



No 4 - BUCKWHEAT OR CULM



No 2 - BUCKWHEAT OR RICE

POWER

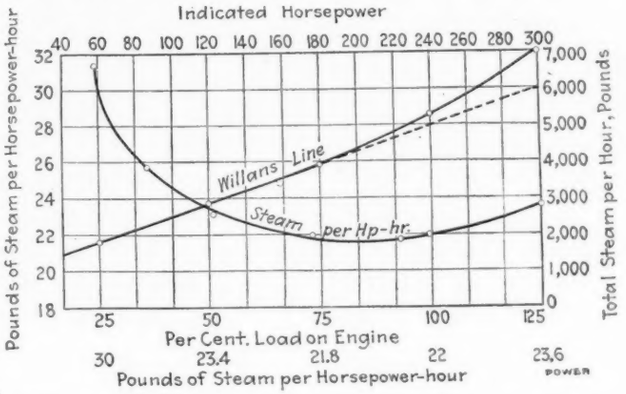
Standards for anthracite-coal sizes as recommended by the American Society of Mechanical Engineers and printed in the Society's *Journal*, November, 1912.

ame	Screen or Opening (Circular) Through or Over Which Coal Will Pass, Inches: Through	Over
Pea	$\frac{7}{8}$	$\frac{9}{16}$
No. 1 buckwheat	$\frac{9}{16}$	$\frac{5}{16}$
No. 2 buckwheat	$\frac{5}{16}$	$\frac{7}{16}$
No. 3 buckwheat	$\frac{7}{16}$	$\frac{3}{32}$
No. 4 buckwheat or culm	$\frac{3}{32}$	..

The sizes specified by the New York City Department of Water Supply, Gas and Electricity are practically the same as the above except for pea coal, which is specified as coal which will pass through ¾-in. and over ½-in. openings.

Load Curves from Ideal Corliss Engines

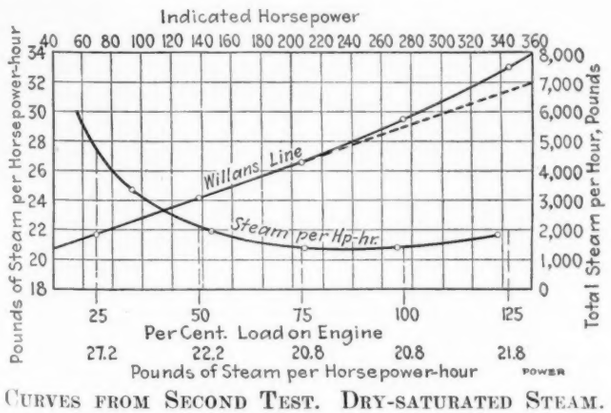
On a recent visit to the shops of A. L. Ide & Sons, at Springfield, Ill., the editor found one of their new four-valve engines upon the testing floor, and was shown the very complete arrangements and apparatus which they have for determining the consumption of their engines. A request for some of their test results has brought the accompanying curves, which are full of interest, showing



CURVES FROM FIRST TEST. DRY-SATURATED STEAM. 124 LB.

the low consumption attained by the use of superheat and the sustained efficiency of the engine through a wide range of load.

The engine is of the high-speed, four-valve type and was described in POWER of July 2, on page 33. The tests from which the curves reproduced herewith were plotted, were made upon a 16x22-in. engine running at 200 r.p.m. exhausting into the atmosphere. The first test was made on Mar. 26 of this year with dry-saturated steam of 124-lb. gage at the throttle. The minimum consumption is 21.7 lb. at 200 hp., or 83⅓ per cent. of the full load.



CURVES FROM SECOND TEST. DRY-SATURATED STEAM. 150 LB.

The total amounts of steam required at various loads are as follows:

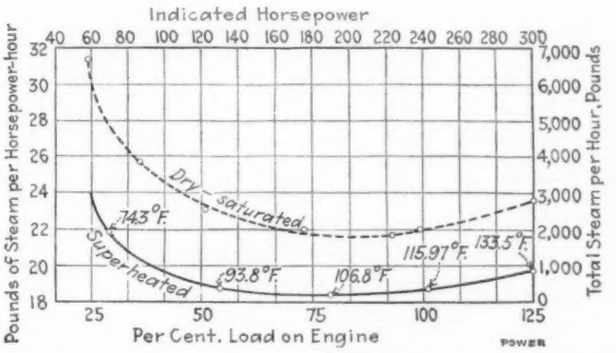
Load per cent. of rating	Load hp.		Steam lb. per hp.-hr.		Total steam lb. per hour
25	60	×	30	=	1800
50	120	×	23.4	=	2808
75	180	×	21.8	=	3924
100	240	×	22.0	=	5280
125	300	×	23.6	=	7080

If these values be set off upon the ordinates corresponding to the respective loads to the scale at the right, and the points connected, the Willan's line, or line of total steam consumption, will be produced. This line runs straight up to about the point of least steam consumption per horsepower but rises as the efficiency falls off with the overload. The best result, 21.7 lb. per i.hp.-hr., corresponds with a Rankine efficiency of 71.8 per cent. and a thermal efficiency, charging the engine with heat above that which it could furnish to the feed water, of 11.6 per cent.

The test from which the second curve was plotted was made upon the same engine on May 7 under the same conditions except that the steam pressure was 150-lb. gage, 25 lb. higher than in the test just considered. With this pressure the rated capacity of the engine is 275 hp., and its steam consumption at various loads was as follows:

Load per cent. of rating	Load hp.		Steam lb. per hp.-hr.		Total steam lb. per hr.
25	68.75	×	27.2	=	1870
50	137.5	×	22.2	=	3052
75	206.25	×	20.8	=	4290
100	275	×	20.8	=	5720
125	343.75	×	21.8	=	7494

The Willan's line has the same characteristics as be-



CURVES FROM THIRD TEST, 125-LB. STEAM, SUPERHEAT 74 TO 133 DEG.

fore. The best steam consumption is 20.7 lb. at about 240 hp. or 87.3 per cent. of the rated load. This is a Rankine efficiency of 70.17 per cent. and a thermal efficiency of 12.11 per cent.

The test from which the third curve was plotted was made with steam of 125 lb. pressure with superheat varying from 74 deg. F. at one-quarter load to 133 deg. at 25 per cent. overload. The lowest steam rate was 18.3 lb. per hp.-hr., at about 190 hp. This represents a Rankine efficiency of 78.7 per cent. and a thermal efficiency of 13 per cent. The curve of the test with saturated steam at the same pressure is indicated upon this chart by a dotted line.

One of the largest dams in Europe was officially put in service on Nov. 16 in the presence of the German Emperor. The dam spans the Bober valley in the Prussian part of Silesia and forms with the dam at Marklissa the backbone of the high-water protection project planned in 1900. It is 920 ft. long at the crest and 460 ft. at the base, 200 ft. high on an average, 24½ ft. thick at the crest and 164 ft. at the base. Cyclopean masonry amounting to 332,000 cu.yd. was used in the construction. The capacity of the reservoir is 1,760,000,000 cu.ft. and the area of the surface is 600 acres, forming a lake 5¼ miles long. Together with the necessary real estate purchases the cost of the dam was $2,450,000.

ELECTRICAL DEPARTMENT

Motors and Motor Applications—II.

By A. B. Morrison, Jr.

Variable-speed Induction Motors

The great disadvantage of the alternating current as compared with the direct current is the inability to obtain a simple motor which will give constant horsepower with variable speed. In general, there are two types of variable-speed motors; one is constant horsepower variable speed, or, to put it another way, a variable-torque variable-speed motor; the other type is constant-torque variable-speed motor in which the horsepower is proportional to the speed. There is no alternating-current motor in which the first class of speed control can be obtained.

There are several methods by which the speeds of induction motors may be varied. These are, briefly: Varying the frequency, changing number of poles, varying the rotor resistance, using two or more motors connected in concatenation and varying the applied voltage. Of all these methods, only two are commonly used in ordinary motor applications; these are: Changing number of poles and varying the rotor resistance. The others are applicable only in very special cases.

The speed of an induction motor is equal to

$$\frac{60 \times \textit{frequency in cycles per second}}{2 \times \textit{number of poles}}$$

From this it is evident that the speed can be decreased by increasing the number of pairs of poles. This is accomplished by using either one winding with a controller so arranged that the connections can be changed to give one-half or double the number of poles or, where several speeds are required, by using separate windings with a controller arranged as described. Usually the squirrel-cage type is used for this purpose, because it is necessary then to change only the stator connections. A wound rotor can be used, but in this case it is necessary to change the rotor connections also. Usually multi-speed induction motors are designed for full and one-half speed or in some cases, for four speeds as 1800, 1200, 900 and 600 r.p.m. The last speed range mentioned is not practicable for anything but squirrel-cage motors.

A motor in which the speed is altered by changing the number of poles is essentially a constant-torque motor and the horsepower is proportional to the speed. At any given speed the operation of the motor is stable; that is, it will operate at a constant speed dependent on the number of poles and independent of the load, barring, of course, the slight drop in speed from no load to full load. The speed changes occur in steps, however, and not gradually; for instance, with the motor operating at a synchronous speed of 1800 r.p.m., the next lower speed is 1200 r.p.m., and there is no intermediate speed. With a wound-rotor motor it would be possible to use a resistance for getting speed variations between any two fixed speeds, but the arrangement would be cumbersome and the speed control not suited to many classes of work.

The disadvantages of the pole-changing method of control are the great number of connections required if the speed variation is to be greater than 2 to 1, the necessity of opening and closing the primary circuit every time the speed is changed, and the decreased power factor at the lower speed. The efficiency of changeable pole motors is approximately the same at all speeds. Also, the maximum and starting torques are approximately the same as for standard motors of the same size and type.

By using a wound-rotor motor with a resistance which can be adjusted by means of a controller the speed can be adjusted below normal. Such a motor gives a constant torque at any given speed and load; hence the horsepower is directly proportional to the speed. The speed, however, is very unstable and a slight change in load will cause a great change in the former. For certain classes of work, such as fans, where the load is constant for any given speed this type of motor can be used provided efficiency is of no particular importance. Its action is very similar to that of a direct-current shunt motor with a resistance in series with the armature. The efficiency at any speed below normal is low because the motor draws full load current from the line and the current not needed actually to perform the work is dissipated as heat in the resistance. Furthermore, the resistance is bulky and occupies considerable space. Where it is necessary to run the motor for short periods only at reduced speeds as for crane or similar work, this type of motor is very satisfactory. Its greatest application is for hoists of various sorts.

Synchronous Motors

Aside from the induction motor the only other type of alternating-current multiphase motor much used in general work is the synchronous motor. This is used, however, only in large sizes and for special applications. Its starting torque is low, but in large sizes and slow speeds particularly it possesses marked advantages over the induction motor, because of its corrective effect on a lagging power factor.

The synchronous motor is practically a revolving field alternator operating as a motor. It can be started by means of a starting compensator, the same as a squirrel-cage induction motor, or by means of a small induction motor. In the latter case the synchronous motor is brought above speed, the induction motor is cut out, and the synchronous motor is allowed to drop in speed until in synchronism with the line when the switch is closed. By over-exciting the fields a leading current can be obtained and this leading current helps to compensate for the lagging current present when induction motors are operated.

When supplied with some type of damping device a synchronous motor will have a starting torque, when started by means of suitable starting compensator, of from 20 to 40 per cent. of full load torque and a "pull in" torque (torque at the moment the machine reaches synchronism) of 15 to 35 per cent. The maximum torque

is about twice the full load torque. Due to the difficulty of starting without drawing excessive current from the line the synchronous motor should be used only where it has to be started and stopped infrequently. If the starting torque is heavy, the motor should be connected to the shafting by a clutch. It should also be remembered that a synchronous motor does not indicate being overloaded by slowing down, but if loaded beyond its maximum capacity it stops suddenly. Also, its speed at any load is constant, dependent only on line frequency.

Single-phase Motors

The use of single-phase motors is confined, for the most part, to sizes below 10 hp. There are several reasons for this. While there are several forms of single-phase motors on the market, for present purposes, they can be divided into two general classes, induction and compensated repulsion.

The induction single-phase motor is very similar, in appearance, to the ordinary multiphase-induction motor. They differ in their method of starting, but when once up to speed their action is the same. The stator winding is very similar to that of a three-phase induction motor and the rotor is usually of the squirrel-cage type or a modification of this, although it may be of the wound-rotor type.

Single-phase induction motors possess no starting torque. To bring them up to speed two general methods are used; namely, phase splitting and repulsion starting. In the former method two windings are used in the stator of the motor; one of these is the working winding, the other the starting winding. These are supplied from the single-phase source and the current in the starting winding is caused to lag behind that in the working winding by means of reactance. This sets up an imperfect rotating field, which is sufficient to bring the motor up to speed under light load, after which the starting winding is usually cut out. There are several different ways in which this "phase splitting" is accomplished. In some cases an external starting box is employed to secure the necessary phase difference in the current, in others the reactance is part of the secondary winding itself.

Where a single-phase induction motor is started by the "repulsion" method the rotor is similar to the armature of a direct-current motor, being provided with form wound coils and a commutator. There are two sets of brushes, bearing on the commutator, these sets being short-circuited upon each other. The stator is supplied with single-phase current, and there is no electrical connection between the stator and the rotor. The currents in the stator set up a flux which reacts on the rotor, repelling the successive coils and thereby causing rotary motion. When the motor approaches synchronous speed a centrifugal device of some description short-circuits the commutator bars and lifts the brushes, transforming the motor into the induction type with practically a squirrel-cage rotor.

The single-phase induction motor, with phase splitting starting device, is suitable for machines in which the starting torque is not over 150 per cent. of full load torque. Almost invariably some type of clutch is used which allows the motor to attain nearly synchronous speed before picking up the load. The starting current with 150 per cent. of full load torque is approximately 250 per cent. of full load current, and the maximum torque is from 150 to 200 per cent. of the full load torque. The speed regulation from no load to full load is good, being better than in the multiphase motor. In general, however, the efficiency, power factor and maximum torque are not as good as in corresponding multiphase motors. They are suited only for driving machinery where the starting torque required is light.

The single-phase induction motor with the repulsion method of starting has a starting torque of from 2 to 2½ times full load torque, with 2 to 2½ times full load current.

The second general type of single-phase motor mentioned is the compensated repulsion motor, in which the line current passes through the stator and also through the rotor by means of two sets of brushes bearing on the commutator. There is also a second set of brushes set at an angle to the first which are short-circuited on themselves. This motor differs from the straight repulsion type in that it contains two additional sets of brushes and the stator and rotor are in electrical contact.

The compensated repulsion motor has a starting torque of 2½ to 3 times full load torque with approximately twice full load current, and the maximum torque is from 3 to 3½ times full load torque. The power factor is very high at all loads, but the efficiency is lower than in the induction motor. This type of motor is well adapted for loads where heavy starting torque is required with sudden overloads. It has the disadvantages of having a commutator and is somewhat more noisy than the induction motor after the latter is up to speed.

If a variable-speed single-phase motor is required, some form of compensated repulsion motor is generally used. The behavior of the motor is very similar to that of the variable-speed wound-rotor multiphase-induction motor with resistance in series with the rotor. It is consequently, owing to its unstable speed characteristics, suited only to such applications as require a steady horsepower at given speeds. Its characteristics as regards starting torque, etc., are unchanged when used for variable speeds. The resistance is inserted in series with the brushes which are normally short-circuited and the insertion of additional resistance decreases the speed. By the insertion of resistance in series with the brushes carrying the line current it is also possible to raise the speed of the motor slightly above synchronism.

[Direct-current motors will be considered in a future article.—Editor.]

Experience in Substation Wiring

By Thomas G. Thurston

Three large 60-cycle transformers were installed to take the place of four motor-generators in a substation which the writer was operating. Up to the time of installation only 25-cycle current had been generated, mostly for rotary converter substations, current for 60-cycle, 4000-volt distribution for the outlying districts being furnished by motor-generator sets each consisting of a 25-cycle synchronous motor direct connected to a 60-cycle, 4150-volt generator. As the 60-cycle system had grown to such proportions as to warrant the installation of a separate generating and transmission system, transformers were installed in the substations and the motor generators were used for emergency and peak-load service.

Shutting down the motor generators rendered the running of the exciters for field excitation unnecessary and

the highest voltage exciter was taken out and sent to another substation.

All the high-tension switches were remote controlled, solenoid- or motor-operated by direct current supplied from the operating busbars which also supplied current for the signal lights, overload relays, etc. These busbars was connected to the exciter busbars when the exciters were running and in case of emergency could be thrown onto a 60-cell, 100-amp.-hr. storage battery. The switchboard lights were connected the same way.

When the exciters were shut down the first problem was to furnish a source of supply for the switchboard and pilot lights. Of course, they could have been carried on

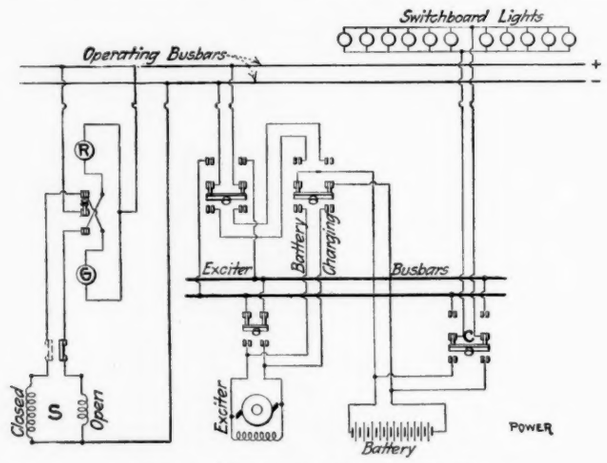


FIG. 1. ORIGINAL CONNECTIONS

the battery, but this was not advisable for several reasons, chief among which was the frequent charging of the battery, thus made necessary. This would have required running a 75-kw. exciter under practically no load for several hours every day, added to this was the increased maintenance and depreciation and risk of the battery being out of service in an emergency.

The operating busbars, of course, had to be supplied from the battery which was the only source of direct current, but this was comparatively insignificant as compared to the current required for the switchboard and signal lights. The original connections are shown in Fig. 1.

After the change a 10-kw., 2200- to 110-volt transformer was connected as shown in Fig. 2 to supply current for the lights when the exciters were shut down. A third wire *A* was run parallel to the operating busbars. The positive connections of the signal lights *R* and *G* and the top clips of the switch *C* for the switchboard lights were disconnected from the operating and exciter busbars respectively and connected to this third wire. The negative top clip of the switch *C* was disconnected from the exciter busbar and connected to the negative operating busbar.

The lower clips of the switch *B* were connected to the transformer, the top right-hand clip to the positive operating busbar, and the switch blades to the third wire *A* and the negative operating busbar. By throwing switch *B* down and switch *C* up the signal lights and the switchboard lights were connected to the transformer. When switch *B* was thrown up they were carried on the operating busbars. This was generally done when the exciters were running and the operating busbars were fed through the exciter busbars through switches *J* and *L*.

The next problem was to provide some means of charging the battery. The high-voltage exciter had been used for this purpose and when it was taken out we were unable to charge the battery with all cells in series by means of any of the remaining exciters. To overcome this difficulty the battery was cut in two and connected to a double-pole, double-throw switch *E*, as shown. In charging, the switch *E* is down and when in service it is up. Throwing the switch down connects the two sections of the battery in parallel, requiring a little over half the normal voltage of the battery to charge it. Throwing the switch up connects the two sections in series.

Everything went along nicely until one night the transmission line to the substation broke down and the room was suddenly thrown into darkness. After some groping an attendant finally located the switch for the switchboard lights and threw it onto the battery circuit. To prevent a recurrence of this trouble a 2-cp. lamp was connected across the battery terminals, as shown at *D*, Fig. 2. This was screwed in when darkness came on and out at daylight.

A surprising complication appeared when the signal lights were carried on the transformer. Referring to Fig. 2, *R* and *G* are respectively red and green lights, red indicating the switch *S* closed and green open, *H* and *I* being the opening and closing coils of the solenoid-operated switch *S*; *F* is the operating switch. Everything was normal until *F* was closed, when the lights would both burn and the one that should have been out would burn with about double its normal brilliancy. For illustration, suppose it was desired to close the switch *S*, it being indicated in the open position. The switch *F* would be closed up. Immediately the light *G* would double

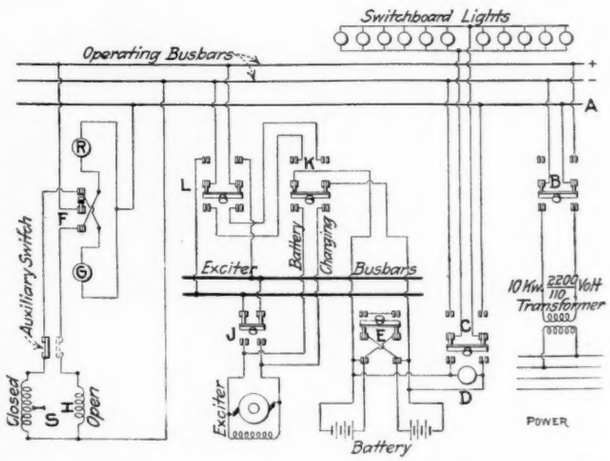


FIG. 2. CONNECTIONS AFTER INSTALLING TRANSFORMER

its brilliancy; when *S* would operate and close, *R* would light as it should, but *G* would remain, showing with double brilliancy when it should have been out, until *F* was opened when it would go out. If *F* was closed down with *S* in the position indicated the red light would appear with double brilliancy and *G* show normal. This held good in either position of *S*, closing *F* either way would show both lights and the one that should be out, with double brilliancy; testing across the switch *F* from the blade to either clip showed approximately 170 volts; the transformer voltage was 110, the battery 120. Where did the excess voltage come from and why did the two lights burn?

GAS POWER DEPARTMENT

Gas Engine Jacket Circulating Systems

By A. C. Wilson

Systems of jacket-water circulation fall into three general classes: gravity, pump and thermo-siphon. The term used in each case to describe the system refers to the force which causes the flow.

If there is a natural head of water available and the quantity is such that it is not necessary to economize, a very simple system may be employed. An inlet pipe is led to the bottom of the jacket and an outlet from the top which is carried to the nearest available drain. A regulating cock is required which should be placed on the outlet pipe and it is important that the outflow be visible, partly so that the temperature may be noted easily, but principally so that the jacket cannot run dry from any cause without the fact being observed. Fig. 1 shows such a system where *a* is the inlet pipe, *b* the outlet, *c* a stop or regulating cock, and *d* a tundish into which *b* delivers.

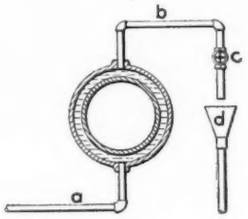


FIG. 1

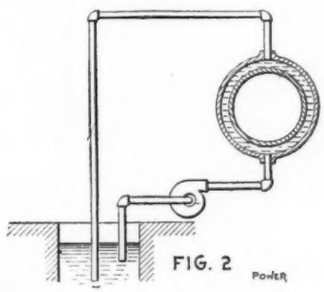


FIG. 2

Circulating systems in which a pump is used may be arranged in a variety of ways; the principal thing to look after is provision for the circulation being maintained or, at any rate, not being altogether stopped should the pump fail.

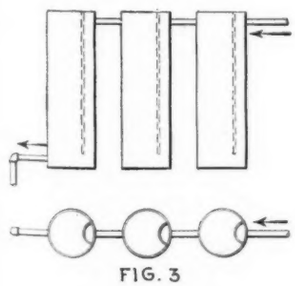


FIG. 3

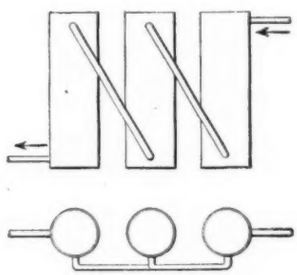


FIG. 4

Assuming the source of supply to be a large overhead tank, a gravity system may be installed, as Fig. 1, with the addition of a pump to return the water to the tank. In the event of the pump failing there would still be several hours' supply available in the tank until the pump has been repaired.

In Fig. 2 the source of supply is at a lower level than the engine and a pump is used to pump the water directly through the jacket. This arrangement is faulty because in the event of anything happening to the pump the circulation is at once stopped. A better arrangement here would be to fit an overhead tank and pump the water direct to the tank, allowing it to flow by gravity through the jacket.

Hard or sedimentary water causes trouble in circulating systems, especially where fresh water is being continually used, as it chokes the pipe connections and puts down scale on the liner. If there is plenty of water available, however, the jacket can generally be kept below the temperature at which the lime salts are deposited, although there will always be a certain amount of scale. To keep the outer surface of the liner at too low a temperature introduces other troubles, such as a cracked liner or seizing of the piston.

In towns, where the cost of water would be a considerable item if run to waste continuously, cooling tanks are commonly used and unless there is some special reason for using a pump, a natural circulation is obtained, due to the difference in density between the hot water leaving the jacket and the relatively cold water entering it; this arrangement is sometimes called a thermo-siphon. As the driving force is that due to the difference in weight of two columns of water the tanks must be placed well above the cylinder or only a very small force can be obtained and in consequence the circulation will be slow.

Tank circulation has several advantages over any other system, among which may be mentioned:

Economy in water.

No power required for pumps.

Circulation automatic and not liable to be stopped by failure of a pump.

The same water being used over and over again, there is less deposit put down if the supply originally contains much lime.

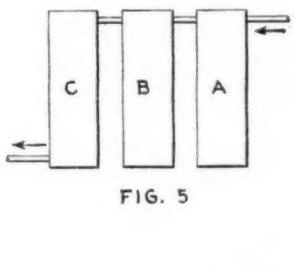


FIG. 5

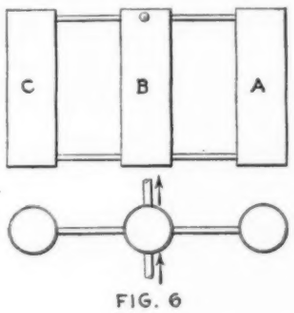


FIG. 6

The number and size of tanks necessary to cool the water depends on their location. Because of the risk of freezing it is not usual to place the tanks outside, but they should be in such a position as to get a good current of air about them. About 30 gal. per horsepower is usually sufficient if the tanks are arranged in such a way as to insure proper circulation.

Figs. 3, 4, 5 and 6 show different methods of connecting circulating tanks which the writer has met with. Figs. 3 and 4 are quite satisfactory as no short-circuiting is possible with either of these arrangements. In Fig. 5 the pockets have been omitted and in consequence water will short-circuit tanks *A* and *B*, and in Fig. 6, where both the flow and return are connected to the center tank, most of the water in circulation would never find its way into the side tanks at all.

If the water level is allowed to fall below the mouth of the return pipe, circulation is stopped and it is a wise

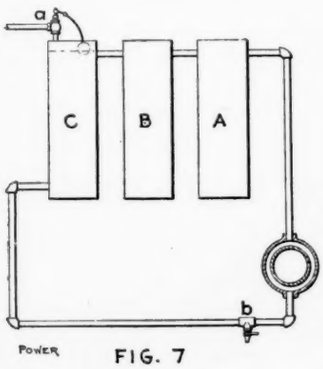


FIG. 7

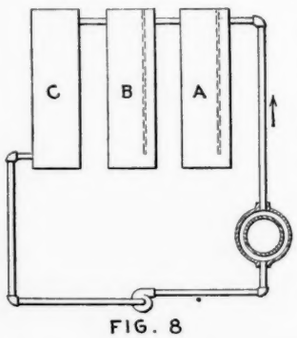


FIG. 8

precaution to connect the makeup supply through a ball tap, although many people rely on the attendant watching this and filling the tank with a bucket from time to time as required.

Should the tanks become too hot, it is the practice of some engineers to open a waste tap in the flow pipe or jacket and let a little water run continually to waste, the ball tap making up the deficiency. This, however, is a doubtful expedient, as it may very easily stop the circulation of the water altogether or perhaps reverse its direction.

When circulating in the usual way the water does not stand at the same level in each tank, being highest in tank *A*, and lowest in tank *C*. If the ball tap is continually running the level in tank *C* is raised, and the circulation in consequence may be stopped, or if the flow from the ball tap is sufficient it may be reversed. If the circulation is reversed the jacket will get the water which is flowing from the ball tap and the system becomes a simple gravity system, but if the level in tank 3 is not raised sufficiently to reverse the circulation, it is quite possible for the jacket to be overheated while cold water runs in at *a* and out at the bottom of the system from the drain tap *b*, Fig. 7.

Sometimes a pump is placed in the flow pipe, as in Fig. 8, to assist the circulation, but when this is done the capacity of the pump must be carefully regulated to suit the rate of flow between the tanks.

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Governor Valves for Gasoline Engines

By L. J. Watson

The practice of using butterfly valves, actuated by some sort of automatic mechanism still continues to quite an extent with gasoline engines wherein the speed is automatically governed by throttling the mixture.

Just why this style of valve is still used for this service the writer is unable to say, for when its efficiency is compared to that of the twin-piston valve, Fig. 1, it is plain that the butterfly valve requires a much greater throw of the operating rod in order to effect the same opening or closure.

To make this point clear compare a 2-in. butterfly valve with a twin-piston valve of the same diameter, each being connected to a 2-in. supply pipe and actuated by the same type of controlling mechanism. A twin-piston valve 2 in. in diameter has a circumference of a little more than 6 in.; therefore each piston commands a port 6 in. long, making a total port length of approximately 12 in. Both ports openings at the same time, it would require the valve to be moved only ¼ in. from the closed position to produce an opening equal to that of the 2-in. supply pipe.

Now consider the opening produced by a butterfly valve (*F*, Fig. 2), 2 in. in diameter and actuated by a crank arm 1 in. long. A ¼-in. throw at the end of the crank arm by the governor rod will move the wings of this valve ¼ in. at two points of its circumference only, and this movement will be mostly lengthwise of the pipe; whereas, to effect opening, it should be in a direction crosswise of the pipe.

Of course, when the valve has been turned on its axis to an angle of 90 deg. from the closed position, the movement at the point of the wings will then be crosswise of the pipe; but the crank arm will have to move through the same angle, which would require a movement from the operating rod of about 1⅜ in., or, over five times that

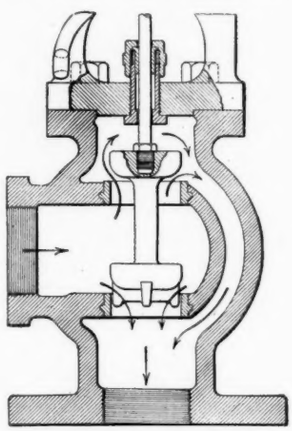


FIG. 1. TWIN PISTON VALVE

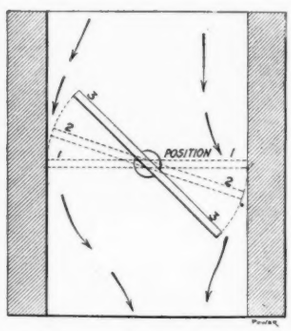


FIG. 2. BUTTERFLY VALVE

for the twin-piston valve. The opening then would not be fully equal to that of the pipe, as the thickness of the valve occupies a small portion of the space.

Furthermore, the most usual position of the butterfly-valve operation is at, or in the neighborhood of, 45 deg. from the full open or closed position, as shown in Fig. 2, position 3. From this the closing and opening effect produced by a given movement of the operating rod is much less than that produced with the twin-piston valve.

The twin-piston valve is balanced; the butterfly valve is not, and as a result, even in gasoline-engine service, where the pressures controlled never exceed 15 lb. (atmospheric pressure) it will naturally resist movement slightly more than the twin-piston type—a matter of significance when it is considered that the valve is moved by a force produced by disturbing the balance of two equal opposing forces.

HEATING AND VENTILATION

Downward Ventilation in a Rockford School*

With a view to reducing the excessive heat losses frequently found in hot-blast heating and ventilation and to attain a temperature practically uniform in all parts of the room with a noticeable absence of drafts, Clinton E. Beery recently installed a system for the Hall School Building, Rockford, Ill., differing in its principles of operation from the usual hot-blast system.

To control the temperature of the air at the ceiling a coil consisting of three 1¼-in. pipes suspended 20 in.

FIG. 1. ONE OF THE ROOMS EQUIPPED WITH DIFFUSERS AND CEILING COILS

below the ceiling, was extended entirely around each room approximately 5 ft. from the exposed walls and 6 ft. from the inside walls. At a point just below the level of the coils fresh air is introduced into the room through a specially constructed diffuser which breaks the current and directs the air in all directions over the breathing zone. The foul-air exits are at the floor line in the four corners of the room. On arriving at the breathing zone, static pressure forces the air out of the room at the floor line, producing a movement of air downward at a velocity of approximately 2 ft. per min., with no appreciable evidence of lateral drafts or air currents. This condition is not disturbed until below the top of the exits, and even then there is no noticeable air movement over the floor, due to the comparatively small exits drawing on a large volume. Each vent screen in this particular installation contains 1.1 sq.ft. clear area, which equals 4.4 sq.ft. for each room, the cubic contents of the room approximating 10,800 cu.ft.

During January of 1912, room No. 3 was fitted up for experimental work to demonstrate the system. This room was in the northwest corner of the first floor, and consequently was the coldest room in the building. Room No. 8 on the second floor, directly above, was an exact duplicate in size, and results obtained in the two rooms were used as a basis of comparison. Each room is 26x31 ft. with the ceiling 13 ft. 6 in. high. Each has a 100 sq.ft. of glass exposure facing the west and 50 sq.ft. of glass exposure facing the north. During the month a comparative test of temperatures was made in the two rooms. Room No. 8 was supplied with 60 sq.ft. of wall radiation in the northwest corner. The air delivery to each room was 1544 to 1560 cu.ft. per min., measured at the exits. Twelve temperature readings were taken at 5-min. intervals, in each corner of both rooms, and also twelve readings of the temperature of the entering air. The outside temperature at the time of the first reading was 2 deg. below zero, and at the time of the last reading, 3 deg. below zero. An extremely cold west wind estimated at 18 to 20 miles per hour was blowing. The maximum and minimum of the twelve corner readings are given in Table 1, and the temperature of the entering air for the twelve readings in Table 2.

TABLE I.—MAXIMUM AND MINIMUM OF ROOM TEMPERATURES DURING TESTS

Room	Northeast Corner	Northwest Corner	Southeast Corner	Southwest Corner
3	70–67	71–70	70–68	69–67
8	83–80	70–68	69–66	69–64

TABLE 2—TEMPERATURE OF ENTERING AIR DURING TEST

Readings	1	2	3	4	5	6	7	8	9	10	11	12
Room No. 3	74	70	72	73	70.5	73	70	71	73	70	73	69
Room No. 8	68	76	109	76	98	72	80	94	68	97	65	83

TABLE 3—TIME RECORDS ON HEATING ROOM TO 68 DEG. F.

	Initial Temp., deg. F	Rise, Deg. F	Time a.m.	Time Required
Room No. 3	38	30	8:50	1 hr. 20 min.
Room No. 8	42	26	9:25	1 hr. 55 min.

FIG. 2. SHOWING UNIFORM DOWNWARD DISTRIBUTION DURING STEAM TEST

The relative humidity readings in room No. 3 showed 63 per cent., and in room No. 8, 54 per cent. Observations of the relative time required to heat the rooms to 68 deg. F. were taken with the heat turned on at 7:30

*Abstract of paper read before the American Society of Heating and Ventilating Engineers.

a.m. Table 3, giving the results, shows a margin of 35 min. in favor of room No. 3.

No difficulty was encountered in operating room No. 3 with the same apparatus while meeting the requirements of the remainder of the building. With the encouraging results obtained, the Hall School was equipped throughout during the summer of 1912 with the new method and has been in operation since Sept. 1. The building is of brick, with wood floor construction, two stories in height, with basement. It contains five regular school rooms on each of the two floors, with wardrobes, recitation rooms, principal's office and corridors. The contents above the basement comprise 124,500 cu.ft. The mechanical equipment consists of a multivane No. 13 fan, direct connected to a side-crank engine operating at 174 r.p.m. All radiation and Vento blast coils are operated on the Webster system of vacuum heating. Johnson Service temperature regulation with intermediate control is used in each room. All galvanized ducts from the plenum chamber are of regulation design.

Table 4 gives the velocity and delivery records of all

TABLE 4—RECORDS OF VELOCITIES AND DELIVERIES AT VARIOUS EXITS

Room	Northwest Exit	Northeast Exit	Southeast Exit	Southwest Exit	Total Delivery Cu.ft. per Min.
1	270	384	326	400	1518
2	390	394	396	350	1683
3	390	400	450	314	1709
4	332	354	460	446	1751
5	364	360	334	332	1529
6	336	346	345	342	1506
7	352	406	334	350	1586
8	434	428	448	366	1843
9	344	328	356	384	1553
10	412	336	370	282	1590

exits for ten 1-min. readings taken after the system had been put in operation, and following is a summary of the observations which have been made since.

A temperature of 85 deg. at the ceiling has been found to be sufficient as a maximum. The temperature throughout the breathing zone is controlled within a range of 1 deg. F. in all parts of the room. Air never enters the room above 73 or 74 deg. F.

There is no noticeable effect on the conditions of the room caused by the glass exposure, but tests showed a slightly higher rapidity in downward movement of the air several inches in from the glass.

There is an entire absence of drafts or currents, and the cold surface of desks and furniture does not contrast against high temperatures of atmosphere surrounding them, as is often complained of in other systems.

During the week of Dec. 15, tests were conducted by the school authorities to determine the efficiency of the system. The temperature was maintained uniformly at 68 deg. and the humidity at 58 per cent. To demonstrate that no cross-currents existed, live steam was introduced into the room through the diffuser in the ventilating system. The steam distributed equally throughout the room and dropped gradually toward the floor like a blanket until it reached the level of the exits through which it escaped. Another test comprised the suspending from wires of a large number of silk threads 36 in. long, to which were fastened pieces of tissue paper. The absence of any swaying of these pieces of paper showed that there were no cross-currents generated by the ventilating system.

From results already obtained, economy of operation is evident, but previous to the time the paper was presented, no efforts had been made to determine the comparative saving. Complete data on this point will be available at a later date and it will be interesting to learn how the economy of the new system compares with the usual type of installation.

CORRESPONDENCE

Use of Pitot Tube in Air Measurements

In the Feb. 4 number of POWER, Frank L. Busey in an article under the above title devotes one paragraph to the coefficient by which the velocity or velocity head at the center of a duct must be multiplied to give the average velocity or average velocity head over the entire cross-section. Several approximate values of the factor are given, but the indiscriminate use of a figure as high as 0.95 would, in many instances, lead to grave errors in fan testing.

Tests recently conducted on a novel air-heating apparatus made it necessary to determine this factor with considerable precision for a 12½-in. circular duct. A study was made of the data for a 16-in. duct contained in Frank H. Kneeland's paper (referred to by Mr. Busey) with the following results:

Type of Pitot Tube	Average Velocity	Center Velocity	Ratio: Av. Vel. / Center Vel.
U.S. No. 1	18,692	19,410	0.963
Gebhardt No. 1	19,138	20,710	0.924
Gebhardt No. 1	20,918	23,080	0.906
Taylor No. 3	14,374	15,670	0.949
U. S. No. 1	13,924	14,820	0.939
Gebhardt No. 1	14,054	15,590	0.901

Not only were the results discordant as far as establishing a ratio of velocities was concerned, but there was a considerable disagreement between pairs of observations, which were taken to represent similar conditions except for the type of tube used. Therefore, the criticism of Naval Constructor Taylor* in the discussion of Mr. Kneeland's results: "The experimental results given by Mr. Kneeland cannot be regarded acceptable to those who like myself are firm believers in the accuracy of pitot tube methods, provided proper pitot tubes are used" seemed none too severe, and a set of experiments was undertaken to establish the ratio, Average velocity ÷ Center velocity, for working conditions. The results of these experiments were communicated to the American Society of Naval Engineers in November, 1912, in a paper: "The Pitot Tube Applied to the Measurement of Air," and as the same may not be available to all interested, a summary is given herewith.

Two standard Taylor pitot tubes were mounted in a 12½-in. circular duct in such a way as to enable one tube to traverse the horizontal diameter and the other to traverse the vertical diameter. These tubes had independent connections to a gage, which was so arranged with stop cocks as to permit of registering the velocity head from either tube, and the gage was fitted with an additional pair of equalizing cocks which could be opened to the atmosphere between tests in order to check the zero reading of the draft gage. The single divisions on the scale of the draft gage were 0.01 in.; and 0.001 in. could be estimated closely. Half-inch intervals were etched on one of the ⅛-in. pipes connecting to the pitot tube.

*Transactions, A. S. M. E., Vol. 33, 1911, page 1168.

TABLE 1. RESULTS OF TESTS TO DETERMINE RATIO OF AVERAGE VELOCITY TO CENTER VELOCITY

Test No.	Barometer, in. of hg.	Relative humidity, per cent.	Temperature of air, deg. F.	Static air pressure, in. of water	Vapor pressure, in. of hg.	Abs. air pressure, lb. per sq. in.	Density of dry air, lb. per cu. ft.	Density of vapor, lb. per cu. ft.	Density of moist air, lb. per cu. ft.	Velocity Head—inches of water: Center of inlet pipe	On 1-in. dia.	On 2-in. dia.	On 3-in. dia.	On 4-in. dia.	On 5-in. dia.	On 6-in. dia.	On 7-in. dia.	On 8-in. dia.	On 9-in. dia.	On 10-in. dia.	On 11-in. dia.	On 12-in. dia.	$\sqrt{\text{Head}}$ at center	Av. $\sqrt{\text{head}}$	Ratio av. vel. to center vel.	Center vel., ft. per min.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
1	30.13	56.27	46.5	0.66	0.18	14.67	0.0782	0.00036	0.0786	0.0709	0.0709	0.0704	0.0701	0.0693	0.0683	0.0676	0.0674	0.0669	0.0658	0.0631	0.0551	0.0426	0.2663	0.2472	0.9282	1042.0
2	29.61	54.47	43.5	0.74	0.16	14.48	0.0777	0.00027	0.0779	0.1030	0.1023	0.1028	0.1021	0.1010	0.1003	0.0998	0.0989	0.0975	0.0954	0.0922	0.0855	0.0748	0.3209	0.3037	0.9464	1260.8
3	29.98	65.48	44.0	1.06	0.19	14.67	0.0786	0.00028	0.0789	0.1602	0.1593	0.1592	0.1581	0.1566	0.1549	0.1532	0.1508	0.1477	0.1443	0.1397	0.1317	0.1190	0.4003	0.3766	0.9407	1563.2
4	29.96	47.10	46.0	1.50	0.15	14.64	0.0782	0.00024	0.0784	0.2018	0.2010	0.2008	0.1998	0.1969	0.1948	0.1915	0.1870	0.1825	0.1773	0.1702	0.1584	0.1446	0.4492	0.4179	0.9303	1759.9
5	29.92	42.51	50.0	2.32	0.15	14.70	0.0779	0.00023	0.0781	0.3221	0.3215	0.3189	0.3153	0.3101	0.3045	0.2958	0.2900	0.2816	0.2740	0.2636	0.2427	0.2111	0.5675	0.5183	0.9133	2227.8
6	29.88	38.34	51.0	2.84	0.14	14.71	0.0778	0.00021	0.0780	0.3881	0.3867	0.3853	0.3808	0.3733	0.3625	0.3529	0.3436	0.3369	0.3343	0.3278	0.3117	0.2846	0.6230	0.5764	0.9252	2447.6
7	29.88	39.03	56.0	3.70	0.18	14.72	0.0771	0.00021	0.0773	0.5030	0.5033	0.4988	0.4885	0.4753	0.4649	0.4464	0.4303	0.4214	0.4191	0.4069	0.3814	0.3452	0.7092	0.6440	....	....
7-a	30.20	55.35	45.0	3.93	0.17	14.89	0.0796	0.00022	0.0799	0.5101	0.5068	0.5023	0.4979	0.4856	0.4734	0.4577	0.4456	0.4375	0.4275	0.4215	0.4033	0.3552	0.7142	0.6541	0.9132	2769.9
8	30.20	54.93	45.0	4.05	0.16	14.90	0.0797	0.00023	0.0799	0.5400	0.5411	0.5364	0.5300	0.5265	0.5070	0.4870	0.4771	0.4658	0.4589	0.4496	0.4264	0.3931	0.7349	0.6773	....	....
8-a	30.01	79.00	47.8	3.77	0.25	14.75	0.0785	0.00040	0.0789	0.5284	0.5251	0.5251	0.5210	0.5116	0.4977	0.4842	0.4716	0.4612	0.4487	0.4295	0.3866	0.3648	0.7269	0.6644	0.9165	2843.6

TABLE 2. PITOT TUBE DATA FROM TESTS BY PROF. C. C. THOMAS

Test No.	Barometer, lb. per sq. in.	Relative humidity, per cent.	Temp. air, deg. F.	Static air press., in. of water	Abs. air press., lb. per sq. in.	Density of dry air, lb. per cu. ft.	Density of vapor, lb. per cu. ft.	Density of moist air, lb. per cu. ft.	Velocity head—inches of water: Center of pipe	On 2-in. dia.	On 4-in. dia.	On 6-in. dia.	On 8-in. dia.	On 10-in. dia.	On 11-in. dia.	On 12-in. dia.	On 13-in. dia.	On 14-in. dia.	On 15-in. dia.	$\sqrt{\text{Head}}$ at center	Av. $\sqrt{\text{head}}$	Ratio av. vel. to center vel.	Av. head in. of water	Center vel., ft. per min.	Lb. of air per hr.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
1	14.2315	63	82	0.430	14.2470	0.07327	0.00167	0.07432	0.150	0.148	0.143	0.138	0.136	0.133	0.131	0.128	0.117	0.100	0.3873	0.3573	0.922	0.1277	1560	8,955
2	14.2315	63	83	0.655	14.2532	0.07314	0.00172	0.07422	0.230	0.228	0.222	0.216	0.210	0.205	0.200	0.190	0.175	0.145	0.4796	0.4408	0.919	0.1943	1932	11,180
3	14.2315	63	84	0.930	14.2651	0.07300	0.00178	0.07412	0.336	0.330	0.320	0.310	0.300	0.294	0.285	0.277	0.255	0.208	0.5797	0.5294	0.913	0.2802	2331	13,200
4	14.2315	63	85	1.085	14.2707	0.07287	0.00183	0.07402	0.385	0.383	0.370	0.362	0.352	0.345	0.335	0.322	0.295	0.240	0.6205	0.5696	0.918	0.3245	2501	14,220
5	14.2315	63	85	1.200	14.2748	0.07287	0.00183	0.07402	0.490	0.455	0.448	0.435	0.425	0.410	0.405	0.390	0.375	0.350	0.6782	0.6184	0.912	0.3824	2729	15,410
6	14.2315	63	87	1.520	14.2863	0.07261	0.00195	0.07384	0.552	0.548	0.538	0.520	0.516	0.500	0.485	0.475	0.450	0.420	0.7430	0.6802	0.915	0.4627	3000	16,990
7	14.2306	63	89.5	1.765	14.2903	0.07220	0.00210	0.07352	0.640	0.632	0.622	0.605	0.589	0.572	0.562	0.538	0.515	0.475	0.8000	0.7273	0.909	0.5289	3234	18,110
8	14.2266	63	91	2.070	14.3013	0.07207	0.00220	0.07345	0.710	0.705	0.685	0.670	0.656	0.627	0.610	0.595	0.575	0.525	0.8426	0.7654	0.908	0.5858	3414	19,080
9	14.2266	63	90	1.150	14.2681	0.07221	0.00213	0.07355	0.385	0.385	0.378	0.365	0.357	0.348	0.340	0.332	0.320	0.292	0.6205	0.5089	0.917	0.3237	2512	14,200
10	14.2266	62	85	0.460	14.2332	0.07287	0.00183	0.07402	0.125	0.125	0.123	0.120	0.117	0.114	0.112	0.109	0.102	0.099	0.080	0.3536	0.3256	0.921	0.1058	1428	8,145

The procedure was as follows: With the fan speed constant a traverse was made with the horizontal tube, a reading taken at each of 23 points across the pipe, and at the same points on the return. The same was followed by a traverse with the vertical tube and each set repeated twice, making a total of 276 observations in each set. The fan speed was then increased and the procedure repeated for the additional points covering the range of fan capacity. The data from the tests are given in Table 1 and the variations of velocity across the cross-section is shown in Fig. 1 in which the ordinates are the square root of the head and the abscissas are the corresponding positions in the duct. From an extrapolation of these curves to a diameter of 12 in. the values in Column 23 of Table 1 are obtained. In arriving at the average square root of the head as tabulated in Column 25, the square root of the head at the extremity of each diameter was given a weight in proportion to the area of the annular ring between the circumferences of circles drawn half-way between the given diameter and the two adjacent diameters. These relative weights were as follows:

Center	0.16
1 in diameter	1.28
2 in. diameter	2.56
3 in. diameter	3.84
4 in. diameter	5.12
5 in. diameter	6.40
6 in. diameter	7.68
7 in. diameter	8.96
8 in. diameter	10.24
9 in. diameter	11.52
10 in. diameter	12.80
11 in. diameter	14.08
12 in. diameter	15.36
Total	100.00

The irregularity in the shape of the curves in Fig. 1 at high velocities is to be charged to the disturbing effect of an elbow in the line. This elbow was eight diameters away from the point at which the pitot-tube readings were taken, but its effect was noticeable at velocities over 2000 ft. per min.

At the same time similar coefficients were computed from some experiments of Prof. C. C. Thomas at the University of Wisconsin and reported by him in a paper "The Measurement of Gases," *Journal* of the Franklin Institute for November, 1911. The primary object of the tests was a comparison of three methods of metering gases: (1) pitot tube; (2) venturi meter; (3) electric meter, but the pitot-tube data, which are given in Table 2, were in sufficient detail to calculate the velocity ratio. In this work pitot-tube readings were taken on one diameter at 1-in. intervals from the center of a 16-in. duct to 5 in. out, and at ½-in. intervals from this point to within ½ in. of the inner surface. The value of the average square root of head was determined by giving a weight to each velocity (square root of velocity head) proportional to the area of the annular ring whose mean diameter was the diameter of the tube circle. This method results in somewhat greater values than those determined by the graphical solution described in the *Journal* of the Franklin Institute, November, 1911.

The variations of the ratios, Average velocity ÷ Center velocity, from both sets of tests is plotted in Fig. 2. The agreement between them is very satisfactory.

It has been suggested that the mean head could be determined by a single tube placed at such a point as to intersect the curves at the mean height. This particular point would be at a distance of 5¼ in. from the center of the 12½-in. duct. There are two serious objections to this method: First, readings on a horizontal and ver-

tical traverse are never the same except at the points nearest the center and it would be impossible to locate the mean point, and, second, the velocity is changing rapidly in this region and the greatest care is necessary in adjusting the tube to this point even if the first difficulty could be overcome. The great advantage of the single central tube with its proper correction factors is brought out by the inspection of the central region of the curve, Fig. 1, which shows clearly that for a ½-in. to either side of the center the velocity head remains practically constant, thereby giving a region in which settings of the central tube may readily be made.

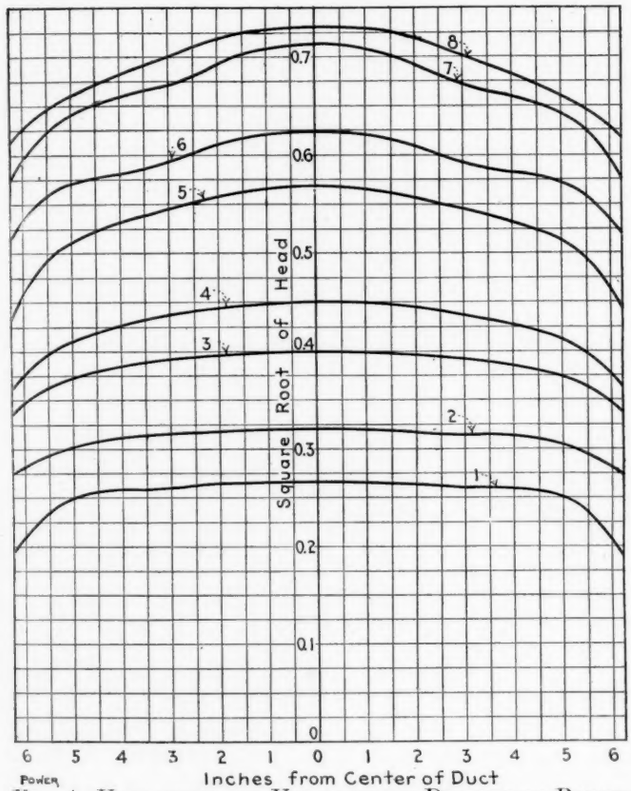


FIG. 1. VARIATIONS IN VELOCITY AT DIFFERENT POINTS ON CROSS-SECTION OF DUCTS

From the plot, Fig. 2, it may be concluded that for ducts from 12 to 16 in. in diameter the ratio, Average velocity ÷ Center velocity, will vary as a linear function of the center velocity from about 0.93 at 1000 ft. per min. to 0.91 at 3500 ft. per min. For smaller ducts the coefficients would probably decrease and for larger ducts a slightly greater value might be expected.

LEO LOEB,
U. S. Naval Engineering Experiment Station.
Annapolis, Md.

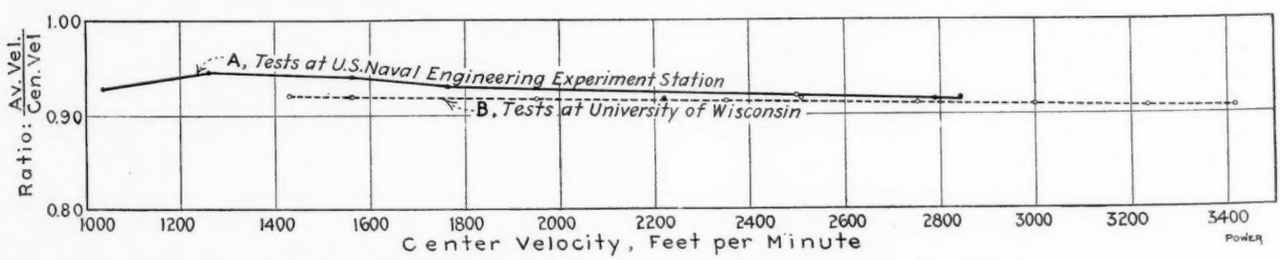

Ratio: Av. Vel. / Cen. Vel.
1.00
0.90
0.80
A, Tests at U.S. Naval Engineering Experiment Station
B, Tests at University of Wisconsin
1000 1200 1400 1600 1800 2000 2200 2400 2600 2800 3000 3200 3400
Center Velocity, Feet per Minute
POWER

FIG. 2. VARIATIONS OF RATIOS OF AVERAGE TO CENTER VELOCITIES

Water Leaves Boiler

Referring to Mr. Walling's inquiry in the Jan. 21 issue and the discussion on this subject in the Feb. 18 issue, my experience with cast-iron sectional boilers leads me to believe that they are subject to a quickly fluctuating water level, as the amount of water they contain is very small. As soon as steam is being raised, the water will drop quickly unless an automatic feeding device is employed. But, in Mr. Walling's case, I think that the principal fault lies in the small return pipes. He says that he has a single-pipe system. The building is three stories high and the main part has ¾- and 1-in. return pipes, enlarged to 1¼ in. at the boiler, while the wing has a 1¼-in. return pipe all the way. Now there is in all probability one or more radiators on each floor from each riser. This means that the ¾-in. pipe has to take care of the condensation from at least three radiators and perhaps six. Even for three, I believe the ¾-in. pipe is too small. It means that water is taken out of the boiler in the form of steam much faster than the small return pipes will allow the condensation to flow back to it. The smallest return pipe in his system, in my opinion, should be 1¼ in. and at least 2 in. at the boiler, and even that is not so large as I should prefer, a 2½-in. pipe would be better.

As for the hammering, if there are no pockets in his horizontal lines and no leaky radiator valves, he will probably find that it will disappear if his return pipes are enlarged. If he has no automatic feed on his boilers, I should strongly recommend that they be installed. They are inexpensive and easily attached and, if they do keep the water level high during the shutdown over night, that water cannot do any harm and could always be blown down. Of course, if the water is going out through the blowoff, he will have to attend to that; but I take it that he means that the water is in the system but not in the boiler. I think that if Mr. Walling will put in larger return pipes most of his troubles with his system will be over.

W. H. PYATT.

Brooklyn, N. Y.

A one-pipe heating system, somewhat resembling the one described by Mr. Walling, gave trouble in much the same manner. The radiators in the wing of the building would get cold and the water leave the boiler. The water in the boiler would rise and fall with a surging effect. A check valve, opening to the boiler, was placed in the return from the radiators in the wing, and after that no more trouble was experienced.

H. G. GIBSON.

Washington, D. C.

EDITORIALS

The Conditions in Massachusetts

Those who are concerned with the procuring and administration of laws for the examination and licensing of engineers and the inspection of boilers, will be interested in the conditions existing in Massachusetts, where this sort of legislation has reached its highest development, and the law and practice of which have been accepted as models.

When the law was adopted some eighteen years ago, the examiner for whom it provided was made an attaché of the Massachusetts district police and the department, although it has increased to some twenty-five inspectors and examiners, has remained a subdepartment of the state police arm, under the authority of the chief of that department. He has no knowledge of engineering, and its members, supposedly versed in all that pertains to the safe construction, installation and use of boilers, engines and power-plant apparatus but totally lacking in police training, are a part of the armed and uniformed force, although they wear their uniforms only upon state occasions and are subject to call only for the suppression of riot, etc.

Subsequent legislation provided for a Board of Boiler Rules, of which the chief inspector of the boiler-inspection department should be chairman and which should be further composed of one member representing the boiler-using interests, one the manufacturing interests, one the boiler-insurance interests and one member who is an operating engineer. These four members are appointed by the governor and do not come under the jurisdiction of the chief of the district police.

In July of last year, the chief of the boiler-inspection department resigned. There had been, in the meantime, a movement afoot in which some of the examiners had been active to divorce the department from its police affiliations. This "pernicious activity" had rendered them *personae non gratae* to the chief of the district police, who sought the advice of the attorney general as to whether the civil-service requirement, governing the appointment of the chief inspector by the governor, still held; for these perniciously active examiners had come out of the previous examination with high ratings. The attorney general found that the necessity of appointing a successor to the chief had never occurred to the "general court"; that no provision was made for such appointment. The governor could not fill the office without warrant of law, because the treasurer's department would fail to find any authority for the emolument. The legal adviser, however, suggested that the chief of the district police had the power to appoint an acting chief who might serve without increase of pay until the legislature could repair the remissions of its predecessor.

This was done, but the Board of Boiler Rules, appointed by his excellency direct, owing no allegiance to the police department, refused to accept his appointee as their chairman and requested the governor to subject candidates for the position to a civil-service examination.

To the civil-service examination came seven of the nineteen inspectors of the department. Perhaps, the others did not want to get mixed up in a strife between superiors; perhaps, they were deterred by other considerations. Of the seven who went in only three got above the sixty-five per cent. necessary to qualify and the chief's repudiated appointee got only fifty-four. The perniciously active came off, we understand, with the highest percentages. But what does that avail when a man has so little sense of discipline and decorum as to go about calling his chief names? The questions asked at the examination are printed on page 292 and are puerile compared with those which some of the examiners are alleged to have proposed to the ordinary garden variety of engineer who appeared before them for examination.

And, so the matter stands. The chief will not withdraw his appointment; the Board of Boiler Rules will not accept him and his excellency says, "Let the general court do it."

We dislike to see the Massachusetts law, or anything connected with its administration, come into ridicule or disrepute. It is an excellent law and we should like to see other states adopt it instead of getting up a variety of dissimilar statutes. Coöperation between the boards of rules appointed under such laws would result in uniformity of practice and the avoidance of different standards for different states. Such inspection is really an exercise of the police power of the state; but so are factory inspection and the activities of the board of health. There seems to be no good reason for linking it up with the criminological branch and making the chief inspector responsible to a police official, on one hand, and subject to the Board of Boiler Rules, on the other.

This mistake may be avoided by other states which adopt the law. Another possible good which may come out of the situation is the realization by some of the inspectors of the ridiculousness of some of the examinations which they give, as indicated by the questions, which are often sent to us as having been asked by Massachusetts examiners, and which lead to the rejection of such men as W. E. Crane, who first applied the double eccentric to a Corliss engine, and failed to satisfy the inspector that he knew enough about its setting and effect to be entrusted with a license to run one.

⁂

Internal Combustion Engine for Central Stations

While with low load factors and cheap fuel, the steam turbine in large units is the only form of heat engine to be considered for central station work, there may be cases where a combination of high load factor and expensive fuel restores the advantage to the internal combustion engine. This is stated by M. Gercke, chief engineer of the Maschinenbaufabrik, Augsburg-Nürnberg, in a paper presented to the Central Station Association at Keil recently, an abstract of which will be found on page 286

There are several stations, he says, in Germany in which Diesel engines and steam turbines are installed together, the engines taking care of the steady load, the turbine of the peaks.

The question suggests itself whether there may not be numerous instances in metropolitan practice where the location of a large internal combustion station in the heart of the system might be advisable. Such a station composed of only enough units, of the greatest capacity available, to take care of the continuous load, would generate current very cheaply, running as it would practically continuously at its rated capacity. The location of the plant in the center of this district to be served would reduce the amount of copper required and whether producers or oil engines were used, there would be freedom from the smoke nuisance, so important in a crowded locality.

It is not beyond the bounds of practicability that the problem of cooling water could be solved by utilizing the waste heat of such a station, in a heating system for the surrounding buildings. The situation of the station in a densely occupied district, would again conduce to this result and the ability to furnish heat as well as current would extend the range of application for the central station service. An economical turbine station on the water front could furnish, through smaller mains than would be otherwise necessary, the surplus current required for the peaks.

The most extensive use for large gas engine plants has been in connection with iron and steel works where gas is practically a byproduct and where the comparatively high price and cumbersomeness of the gas engine counts for most as against the burning of this gas under boilers and the utilization of the steam in the simpler and less expensive turbine. May we not see a revival of the large gas engine installation, under conditions which will offer it a better chance?

New York Boiler Laws

Is New York State to have a boiler-inspection law? Those who listened to the remarks of the speakers at the recent dinner of the American Institute of Steam Boiler Inspectors could not help but feel optimistic in the matter. However, this is not the first time, by any means, that boiler-inspection laws have been proposed for New York. Several times the prospects have seemed particularly bright for the adoption of such laws when opposition was encountered, influential enough through legislative channels, to defeat them.

As to the desirability of a state inspection law, little need be said—its merits are obvious. In fact, they are so overwhelming that its opponents dare not come out in the open. Any opposition that may develop is likely to spring from certain interests in cities within the state, which have their own boiler regulations. Boiler inspection in New York City has long been recognized as deficient. Not that the inspectors are incompetent, but they are laboring under a system which prevents them from doing full justice to their work. It is time for a change from these conditions.

Equally as important as the construction and inspection of a steam boiler is its operation. The best boiler in the hands of an incompetent man is more dangerous than a stick of dynamite. This leads to the consideration of a state license law. One without the other affords only partial protection. In Massachusetts, an engineers' and firemen's license law, a boiler-inspection law and rules governing the construction of boilers are enforced conjunctively. One other state, Ohio, and several cities, have adopted the Massachusetts law as a whole, while other states and cities have adopted it in a modified form, or certain of its provisions. The Massachusetts rules are regarded as the standard, and New York, which boasts of being the Empire State, cannot afford to compromise but should adopt the best there is.

The present movement has the advantage over previous ones in that it is being pushed by the American Institute of Steam Boiler Inspectors, whose judgment in such matters should carry much weight. Furthermore, it has the support of many prominent boiler manufacturers and insurance companies and will no doubt be heartily indorsed by all engineering societies within the state.

May success be the reward of this latest effort.

No More Merchant Pipe

Standard pipe is a good deal thicker, and hence stronger than it need be for many purposes, especially in the smaller diameters. It has been the practice to draw pipe, to order, of less than the standard thickness for use under conditions in which the standard pipe was not necessary, and this pipe, of the same outside diameter as the standard, threaded and coupled like it, similar in every particular except in thickness and weight, has been known as "merchant" pipe.

The reduction in weight per running foot has been kept within limits and the merchant pipe subjected to the same test, seven or eight hundred pounds per square inch, as the standard pipe. So long as the purchaser knew what he was getting and used it intelligently no harm was done, and unnecessary expenditure was avoided.

But with a lot of pipe of less than standard thickness floating around, all looking alike, all threaded alike, only the expert, or he who was particular enough to weigh or gage it, could be certain that he was getting what he paid for, and this thin pipe was likely to be inadvertently perverted to uses for which it was not adapted.

The National Tube Co., therefore, has announced, and we understand that most if not all of the other pipe producers are adopting the same attitude, that it will draw no more merchant pipe, so that when a man buys a length of pipe he may be sure that it is within five per cent. of the prescribed weight above or below. It may look arbitrary to refuse to draw an order of pipe of special thickness (or thinness) for a customer who has use for a large quantity for gas or other low-pressure service, but in clearing the market of undersized stock and making "standard pipe" mean one thing, the move is the best thing which has happened for the purchaser of piping in many years.

According to one engineer the extortionate prices imposed by the coal barons are not undiluted evil. "I wish the stuff cost twice as much," he says, "then a man would get a little more credit for saving it."

READERS WITH SOMETHING TO SAY

Interesting Piping Problem

In power-plant design the end which suffers most from lack of careful study is usually the piping. Much of the piping is often left until the last moment after the position of all the various pieces of apparatus has been decided upon, and is rushed through without much thought as to system or cost. I recently had brought to my attention a piping layout in which a saving of several hundred dollars over the original method was effected by a careful study of the whole problem and a revision of the plans.

Referring to the diagram, A is the power plant and it is desired to carry steam to the point D. BD is a tunnel

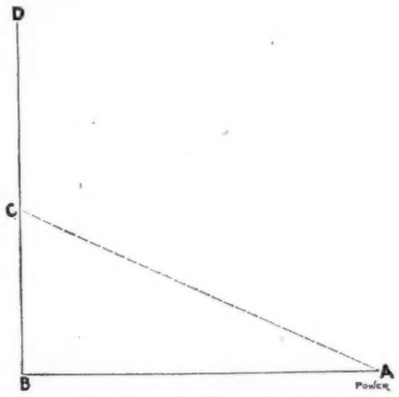


Diagram of Pipeline

in which the pipe is carried on hangers. From A to BD it is necessary to run the pipe in conduit. The original layout was to run the pipe through a conduit to B (ABD being a 90-deg. angle) and then through the tunnel to D. Estimates were secured for installing the pipe in the conduit at $7 a foot and in the tunnel at $3 a foot. If run the way originally intended the cost would be $10,000. If run from A to D in a straight line through a conduit the cost would be $9900. From a study of the layout it was evident that by running the conduit to some point C, intermediate between B and D, the work could be done for the least money. The problem, therefore, was to locate C. This can be done by the use of trigonometry and a trial and error method, but is long and tedious. A much simpler method is the use of what is known as "maxima and minima" in differential calculus.

If we call the distance

$AB = a$;

$BC = x$;

and

m = Cost per foot for a;

n = Cost per foot for x;

then without giving the various steps by which the result is obtained

$$x = \frac{an}{\sqrt{m^2 - n^2}}$$

For the case given BC should be 475 ft. This gives a total cost of $9325.

If there was a pipe going from C the other direction in the tunnel and its cost was z the equation would be

$$x = \left(\frac{n - z}{\sqrt{m^2 - (n - z)^2}}\right) a$$

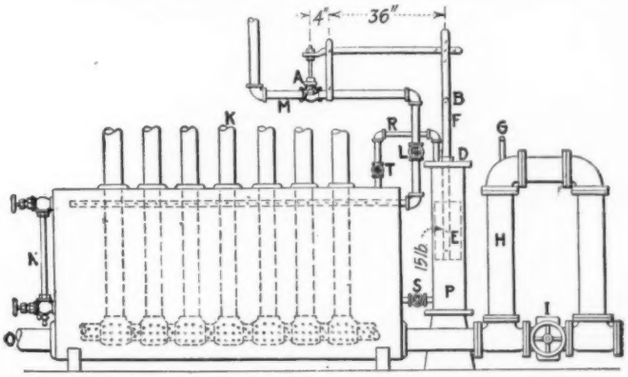
W. L. Durand.

Brooklyn, N. Y.

Receiver and Automatic Control

In the plant where I am employed, a tank 5 ft. in diameter and 12 ft. long receives the exhaust and returns from five engines and two paper machines. The boiler-feed pumps remove the water. Make-up water was always regulated by hand and it was impossible to keep the water level at a normal height; there was also a great variation in the temperature.

A number of float arrangements had been tried to automatically regulate the make-up water, but without success. I tried the device illustrated herewith, which worked satisfactorily. A is a common globe valve with a smooth stem. B is a ball joint. At first we used a piece of 8-in. pipe for the float chamber P, with the top open to the atmosphere, but at times the water in the receiver would get above 212 deg. F. and, owing to the small area of the atmospheric pipe K, the pressure would rise from 1 to 5 lb. pressure, which forced the water in the receiver, up into the float chamber, closing the valve at the time it should be open.



Piping of the Return Tank and Automatic Control

To overcome this we put a cap on the top end of the chamber P with a packing gland at D for the float rod to run through, and connected a pipe R to the top of the receiver and float chamber to equalize the pressure.

A piece of ⅜-in. nickel-plated pipe F was used for the rod. It was soldered to the bottom and top of the float and a ⅛-in. hole drilled at E and another below it. This overcame the trouble encountered from collapsing floats.

There is 25 lb. pressure in the pipe M. Valves S and T

are to shut off the float chamber, *H* is the overflow; *G* is a vent to prevent siphon action; *I* is the blowoff valve; *L* is a valve to be used if the float is out of commission, *N* is a gage-glass; *O* the pump suction, and return pipes are shown on top of the tank.

R. A. Hafford.

Groveton, N. H.

Corroded Boiler Tubes

The boiler tube from which the accompanying illustration was made was lately removed from a horizontal return-tubular boiler, near Detroit, Mich. This boiler is using water containing a considerable quantity of sulphates. The scale is not generally thick nor hard. Raw water, in the winter only, makes up about 50 per cent. of the feed water, the remaining 50 per cent. is condensation from the heating system, which is returned to the boiler by a return trap.

Corroded and Cracked Boiler Tube

Natural gas is used for fuel, together with refuse, consisting principally of wood.

Cutting of the tubes by corrosion on the exterior of the tube occurs close to the inside of the front head. These tubes are from three to about 10 years old. The faulty ones are not all close to one another but scattered about the nest. Other boilers belonging to the same firm, located across the street, have not experienced any such trouble in over ten years.

We fail to find any good reason for the cause of this trouble and thought that perhaps some reader might have had a similar experience. Comments by those who can throw light on the matter will no doubt prove of general interest to engineers and firemen.

J. B. Latour.

Toronto, Ont., Can.

Calculating Scales

The calculating scale, Fig. 1, is for horsepower computations; Fig. 2 is for figuring pipe areas for the flow of steam under different conditions of pressure and velocity. The horsepower calculator is used in checking the horsepower of engines, but the limits are so large that extreme accuracy is not attainable with the size of scale used.

The steam-flow scale is of value for anyone interested in the design or operation of steam machinery. With larger subdivisions the error of about 2 per cent. would be decreased.

Referring to Fig. 1, assume that the horsepower of a 14x24-in. engine is to be found, having a mean effective pressure of 52.5 lb. and a speed of 150 r.p.m. Place the stroke scale *C* opposite the mean effective pressure scale *B*

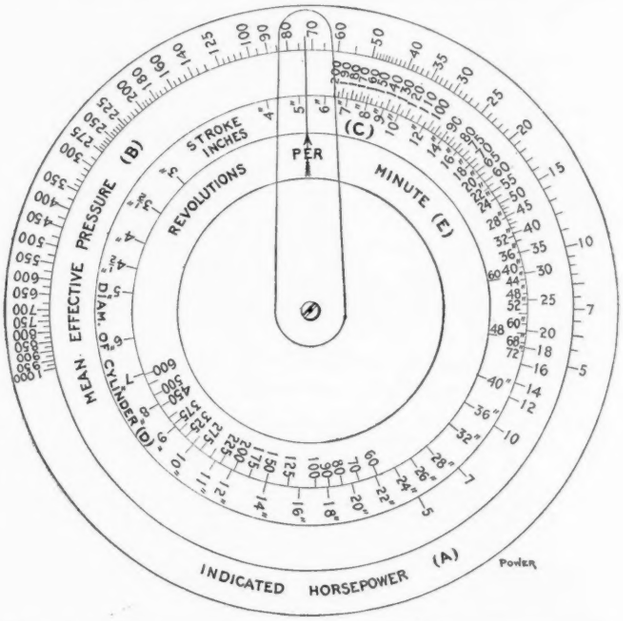


Fig. 1. Chart for Computing Engine Horsepower

and revolve the top disk until the revolution per minute scale *E* is opposite the cylinder-diameter scale *D*. Then revolve the rider arm to the arrow and read the horsepower on the scale *A*. Thus 24 and 52.5 will line up on the scales *C* and *B* and 150 and 14 will line up on scales

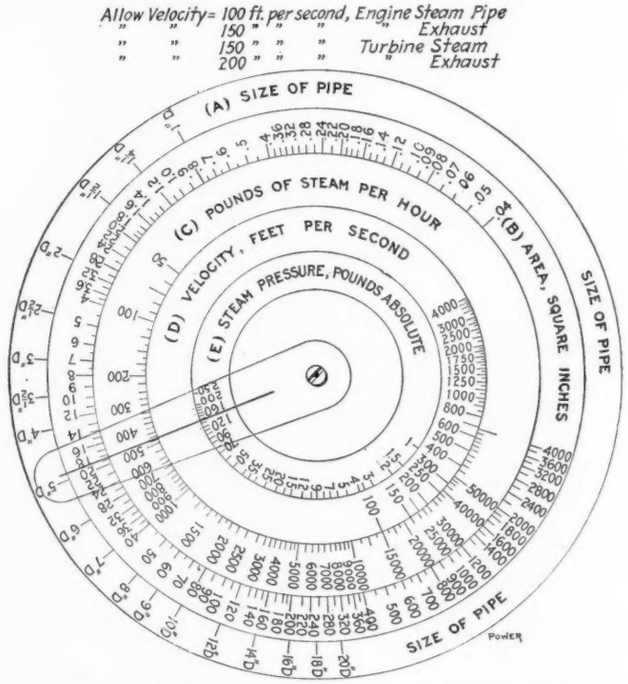

Allow Velocity= 100 ft. per second, Engine Steam Pipe
" " 150 " " " Exhaust
" " 150 " " " Turbine Steam
" " 200 " " " Exhaust
(A) SIZE OF PIPE
(C) POUNDS OF STEAM PER HOUR
(D) VELOCITY, FEET PER SECOND
(E) STEAM PRESSURE, POUNDS ABSOLUTE
(B) AREA, SQUARE INCHES
SIZE OF PIPE
SIZE OF PIPE
POWER

Fig. 2. Chart for Computing Pipe Areas

E and *D*. The line on the rider, when covering the arrow, points to 72.5 hp., or double that for the total horsepower developed per revolution, 145 hp.

The chart shown in Fig. 2 is manipulated similarly. Assume the steam velocity is to be 100 ft. per sec., steam

pressure 150 lb. and that 15,000 lb. of steam is required per hour. Place the velocity scale *D* opposite the desired quantity of steam on chart *C* and move the rider over the steam pressure sidewise on scale *E*, the area will be shown as 18 sq.in. on the scale *B*.

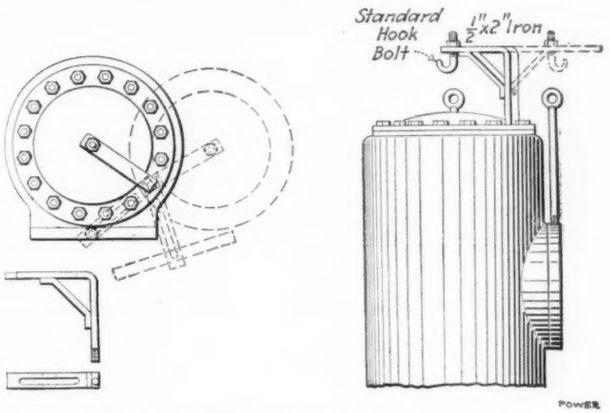
G. H. Bascome.

Elmira, N. Y.

Device for Removing Pump Cylinder Heads

The device illustrated herewith I find very handy for removing water-cylinder heads on horizontal pumping engines where the crane service is available.

The only provisions necessary for applying it are an eye-bolt hole drilled and tapped in each head and the sub-



Device for Removing Cylinder Heads

stitution of a tap bolt for a stud in the holes where the bracket leg screws in. The slotted oblong hole in the bracket, facilitates removing the lower side heads which may be handled either by a long hook bolt or the same hook bolt and a long eye-bolt, preferably the latter, as shown.

This makes a portable outfit for this kind of work, and its removal after using leaves the appearance of the pump unchanged, there is no necessity to hang unsightly timbers, ropes, hoists, etc., overhead.

W. Oliver.

Racine, Wis.

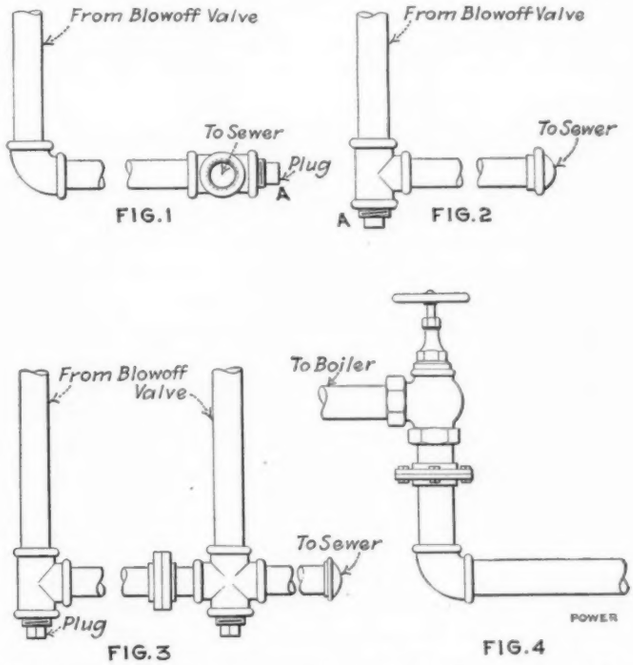
Openings to Detect Leaky Blowoff Valves

It is good practice to have an opening in the blowoff pipe between the blowoff valve and where the water is discharged, as then one can see at once if the blowoff valve is leaking or not.

Especially should this be done where the water is discharged into a closed tank or catch basin. Our blowoff pipe discharges the water into a catch basin outside of the building and the blowoff pipe is arranged as shown in Fig. 1. After I start filling the boiler with water the plug *A* is removed and the end of the tee left open until steam is raised in the boiler. In this way, if the blowoff valve leaks ever so little, it can be detected at once. I also find this affords a good means to make sure that the blowoff valve is not leaking after blowing some of the dirty water out of the boiler in the morning or during the noon hour.

Where there is room the tee could be placed next to the blowoff valve, as shown in Fig. 2, and where the blowoff pipe of two or more boilers discharges into one main trunk pipe leading to a closed tank or catch basin, the piping could be arranged as shown in Fig. 3.

One bad mistake some engineers and steam fitters make in connecting a blowoff valve to a boiler is that they fail to place a flange or union near the valve between it and where the water is discharged. Often the flange or union is placed a long distance from the valve, then, if it is necessary to take out the valve for repairs or to re-



Blowoff-pipe Connections Arranged to Indicate Leaky Valve

place it, a lot of piping must be disconnected, causing unnecessary labor and expense. All this trouble can be avoided by placing the flange or union as near as possible to the blowoff valve.

H. A. Jahnke.

Milwaukee, Wis.

Engine Vibration Tests

In the advertisement of the Rollins engine in a recent issue of Power I read that a Massachusetts engineer balanced a pencil on the top of the cylinder and it stood upright all day.

This certainly speaks highly for the engine, but I once performed a little experiment myself which I thought rather hard to beat. While running one of those 125-hp. engines installed in the F. B. Pierce Chair company's shop at South Keene, N. H., I balanced three ten-cent pieces on various parts of the engine, one on the pillow block, one on top of the cylinder over the back head, and one on the frame over the girder, and they would stand that way for hours or until I got short of spare change.

Arthur H. Parker.

Warren Summit, N. H.

QUESTIONS BEFORE THE HOUSE

Air Receiver Explosions

Some months ago, while investigating the causes of an explosion of an air receiver that was carrying 100 lb. gage pressure regularly, I found and reported that the inside of the receiver was coated with grease to the extent of ¼ in. thick or more. Since that time, I have had opportunity to examine four other compressors and receivers. In each case the interior of the shell was coated with grease from ¼ to ½ in. thick. I immediately began to look into the lubrication of the compressor cylinders and have since been watching the lubrication of various running compressors, with the following deductions, some of which agree with the previous discussion on this subject:

Many mines employ a man to run the compressor and, at the same time, watch the hoist located near-by. This man must set his lubricator to feed the oil and, nine times out of ten, adjusts it to feed about the same as for a steam cylinder. Frequently he starts it cold and after running it awhile goes back and finds a tiny stream of oil running into the cylinder. Again, when the lubricator is empty, he grabs an oil can, as he gets a hoisting signal, and rushes over to put a few drops into the lubricator before answering the signal. A change in engineers frequently means a change in oils and a different rate of feed. Even if the engineer has only the compressors to care for he is generally a "steam-man" and gages his steam and air-cylinder lubrication alike.

Frequently incompetents are entrusted with motor-driven compressors, and are instructed to watch for hot bearings, pull the switch if anything goes wrong and send for the electrician. All that such men can do is to pour out oil, which they generally do.

Many compressors draw their air from the hot engine room and out here on the desert of Nevada, the mine compressors frequently have open unfiltered intakes drawing in desert air with a goodly portion of dust and alkali. Assuming this intake temperature to range from 120 to 160 deg. F. and the compression from 100 to 104 lb. gage pressure, the heat of compression may run up the temperature to 500 deg., which is higher than the flash point of many cylinder oils. The stopping of the discharge by a valve stem sticking, or grease or carbon accumulating to reduce its area, or by running the compressor at overspeed, may cause some of the heated air to come back for recompression and still further advance the temperature. Cylinder jackets might help some, but the use of muddy mine water without any attention would soon reduce their value.

Many steam-driven compressors have throttle governors to reduce the speed when occasion requires, but, in several cases, I have observed that the lubricator was left adjusted for the maximum speed.

To avoid air-receiver explosions, it would seem that several points can be emphasized. Use only the best of compressor oil and use only as little of it as possible, far less than any steam cylinder requires. At the lowest point of the receiver tap for a drain and screw in a short nipple. Then bush the out end of the nipple up to a 3- or 4-in. pipe and put on a piece 6 in. to 1 ft. long. Reduce again to the size of the stop cock valve it is desired to use. This forms a pocket or trap for the moisture and oil, which can be readily blown out. In the absence of a trap, a hole is blown in the residue when the blow-off cock is opened. The little hole immediately fills up and the thick oil deposit does not flow at all. I have seen soft soap and water and potassium or sodium hydrate and water used for cleaning grease and gum from cylinders, air passages and receivers with considerable success. Gasoline, kerosene, naphtha or the like should not be used as they are likely to form highly explosive gases inside the air passages. In any event, a manhole should be provided in the receiver so that it can be readily scraped and cleaned.

Tonopah, Nev. LETSON BALLIET.

Mr. Booth's Condenser Problem

The problem given by W. H. Booth in the Dec. 17 issue relative to the source of power that lifts the cooling water into a jet condenser 17 ft. above a canal, proves to be of unusual interest.

By his wording, I assume that the question means that the outlet from the air pump is open to the atmosphere 17 ft. above the canal, and that from there the water runs down freely through an open pipe or duct to the canal. This assumption being correct the answer to the question is that the lifting power comes directly from the air pump.

Let us imagine a theoretical case of a plant of, say 500 hp., where there are 175 lb. of steam condensed per minute, assuming a ratio of cooling water and steam of 30 to 1. The cooling water would amount to

$$30 \times 175 = 5250 \ lb. \ per \ min.$$

Adding this to the 175 lb. of condensate gives 5425 lb. of water pumped out by the air pump per minute. To make the problem simple, assume there is no air in either the condensed steam, or in the cooling water and that the wet-air pump handles a full cylinder of water every stroke. Assuming, also, that the hotwell temperature is 115 deg. F., the water handled by the wet-air pump would weigh 61.8 lb. per cu.ft. Then we have

$$\frac{5425}{61.8} = 87.8 \ cu.ft.$$

of water pumped per minute.

Further, let the vacuum be about 26 in., giving an absolute condenser pressure of 2 lb. per sq.in. Then the head against which the wet-air pump must discharge is

$$14.7 - 2 = 12.7 \ lb. \ per \ sq.in.$$

Neglecting slippage, friction and all other losses, the power developed in the water cylinder of the air pump would be

$$\frac{87.8 \times 12.7 \times 144}{33{,}000} = 4.86 \ hp.$$

Of this amount, the cooling water represents

$$\frac{5250}{5425} \times 4.86 = 4.71\ hp.$$

Now, supposing a purely theoretical (and really impossible) condition, that the cooling water is as hot as the hotwell water just to have one horsepower balance theoretically the height to which the available vacuum would lift the cooling water. This would be

$$\frac{12.7 \times 144}{61.8} = 29.6\ ft.$$

To lift 5250 lb. of cooling water per minute through 29.6 ft., would require

$$\frac{5250 \times 29.6}{33{,}000} = 4.71\ hp.$$

Thus we see that the power exerted by the suction of the condenser (the atmosphere) is exactly the same as that developed in the wet-air pump, pumping the cooling water only from vacuum to atmosphere. Thus, the power to lift the cooling water comes from the pump.

As a matter of fact, the theoretical condition that the cooling water is as hot as the hotwell water is impossible; so assuming the cooling water at 60 deg. F., it would weigh 62.4 lb. per cu.ft., and the actual column that the vacuum would support would be

$$\frac{12.7 \times 144}{62.4} = 29.3\ ft.$$

and the horsepower necessary to lift this column of water would be

$$\frac{5250 \times 29.3}{33{,}000} = 4.66\ hp.$$

The difference between this and the 4.71 hp. or 0.05 hp. is the little extra burden that the air pump has to bear, due to the heat put into the water with its consequent expansion.

Suppose we have a surface condenser. Without any air, the air pump would develop only the power required to pump the 175 lb. of condensed steam per minute. This would represent a volume of

$$\frac{175}{61.8} = 2.83\ cu.ft.$$

which against 12.7 lb. pressure would be

$$\frac{2.83 \times 12.7 \times 144}{33{,}000} = 0.157\ hp.$$

If there were any lift, there probably would be an equal drop at the outlet; so, except for friction, the circulating pump requires no power at all.

If we have a barometric condenser, without any air pump, the condenser could, theoretically, be placed a distance above the hotwell equal to the height of condensate that the atmosphere will support. This we have already figured to be 29.6 ft. If the cooling water were at the same temperature, this too would be lifted this height by the vacuum (atmosphere) and once started would continue round and round, assuming no friction, without any pumping. However, the cooling water to be of any use must be cold, so assuming 60 deg. as before, the height to which the vacuum would lift this water would be, as already figured, 29.3 ft. The extra lift of 0.3 ft. must be supplied by a circulating pump, as well as considerably more lift necessary to insure that the condensing chamber shall be high enough above the hotwell to free itself quickly of water. This usually is not less than 34 ft. total.

Applying these theoretical considerations to Mr. Booth's problem, we find the lift in his case is only 17 ft., which is much less than the head against which the pump must discharge. The rest of the available column must be absorbed by water friction, or a throttling valve, which is the same thing, so the air pump is really lifting all the cooling water into the condenser and overcoming this friction also. The friction does not throw any more load on the air pump because the pump must develop the full power in getting the cooling water out against the atmosphere and this may be absorbed entirely by lifts or by a shorter lift with more friction. The power is there; so, within lifting range, the height of lift does not really alter the power.

It is perfectly possible to decrease this power by properly piping the condenser outlet back to the canal level. This would then form a siphon and, barring friction and air and the change in volume, due to water temperature, no power would be required to circulate the cooling water. The atmosphere would raise the water against gravity and gravity would return it against the atmosphere. In this case, any power obtained from a water turbine in the return to the canal, as Mr. Booth mentioned, would come from the air pump itself.

If a jet condenser is placed on a level with the source of cooling water supply, the air pump must do the entire work of pumping out against the atmosphere. This is because, there being no lift, the cooling water must be choked back by a valve, lest it swamp the condenser and run back into the engine; and there being no lift there can be no drop on the discharge, so gravity cannot help to remove the water. If, however, the condenser can be elevated so as to use the vacuum to elevate the water and the rest of the vacuum to overcome friction and valve control, the consequent drop at the discharge, less its friction, will be almost enough to get the cooling water up into the condenser. Then the air pump, aside from handling the condensate, need overcome only friction in removing the cooling water. By the elevation the vacuum overcomes gravity, which can return the work, instead of valve friction, which cannot return anything. Some provision might have to be made to get the air down the discharge pipe, but a high-water velocity would probably carry away all the air contained in the column of water.

This brings us to the conclusion that, properly arranged, neglecting friction and air, no kind of condenser should require power to handle cooling water except that to overcome the effect of a slight expansion of the water, due to an increase of temperature. Thus expansion would tend to make the siphon run backward and this tendency must be overcome with a pump. Even in a barometric condenser, where the work seems to be done by gravity, the energy of the steam must be used to lift the steam itself up into the condenser (so many pounds' weight against so many feet lift) for it to run out again by gravity against the pressure of the atmosphere. If the engines are above the condenser, then this lifting is done between the boilers and the engine; or if the boilers too are above, the work is done by the boiler-feed pumps.

R. S. Bayard.

New York City.

INQUIRIES OF GENERAL INTEREST

Grate Area—What should be the grate area of a 200-hp. boiler?

M. P. M.

A 200-hp. boiler should be capable of evaporating.

$$200 \times 34\tfrac{1}{2} = 6900 \text{ lb. of water per hr.}$$

from and at 212 deg. F. at atmospheric pressure. For ordinary good operation 9 lb. of water should be evaporated per pound of coal. Then the coal required per hour is

$$\frac{6900}{9} = 766.6$$

Average practice allows 18½ lb. of coal per hour per square foot of grate. Then

$$\frac{766.6}{18.5} = 41.4 \text{ sq.ft.}$$

or, in round numbers, 42 sq.ft. should be the grate area.

Sliding Foot on Engine—Why does a Rice & Sargent engine have a sliding foot?

S. J.

A sliding foot is placed under the engine to permit the frame to assume its natural alignment without being strained, as it might be, from expansion and contraction, or springing of the engine bed.

Greatest Pressure in a Boiler—What part of a B. & W. boiler is subject to the greatest pressure?

F. F. M.

The greatest pressure per unit of area comes on any boiler at the lowest point that is subjected to the hydrostatic pressure of the water in the boiler. In a B. & W boiler, that point would be the mud drum.

Oil Engine Cylinder Oil—What is a suitable cylinder oil for oil engines of the semi-Diesel type?

J. L. L.

A light, clean mineral oil should be used, having a flash point of at least 370 deg. F. and a specific gravity of from 0.913 to 0.907 or 24 to 25 deg. on the Baumé scale.

Boiler Circulating Pipes—Where is the circulating pipe attached in a vertical shell steam boiler and what is its particular function?

J. S.

Circulating pipes are sometimes placed in vertical boilers connecting the water space above the fire-tube sheet to the lower part of the water leg, to induce a rapid circulation in the water leg to prevent mud and sediment accumulating there.

Furnace Draft Velocity—How can the velocity of the draft in a boiler furnace can be calculated from the force of the draft?

J. B.

The force of the draft cannot be converted into the velocity of the draft, because the obstructions offered by the grate and fuel bed and the friction of the air through the passages cannot be told. The velocity of the draft can be ascertained by placing an anemometer in the air entrance or draft passages in the flues.

Concrete Boiler Setting—Is concrete a satisfactory material for a boiler setting or can it be made so by lining with fire brick?

J. G. B.

Brick is better for the side walls of a boiler setting than concrete, even when the latter is lined with fire brick and an air space is left, because expansion of the concrete would be inevitable and the walls would be very likely to crack in such a way as to be weakened and to admit air through the crevices formed, and this would be seriously detrimental to the boiler economy.

Boiler Horsepower—What is meant by the equivalent evaporation of a boiler and how is the boiler horsepower rated?

W. N. M.

The equivalent evaporation is the number of pounds of water that would be converted into steam at 212 deg. F. and atmospheric pressure by the same amount of heat that was utilized in the actual evaporation; i.e., the weight of water converted into steam at the pressure and temperature obtaining. A boiler horsepower is usually rated as each 34½ lb. of equivalent evaporation per hour.

Hydraulic Elevators—Explain the principles of "Standard Plunger" and high-speed hydraulic elevators. What is the cycle of operation? Can the same water be used indefinitely in the operation? Is the use of air necessary?

C. R.

Standard Plunger and other high-speed hydraulic elevators are operated by the pressure of water maintained in an accumulator or in an elevated tank. When the water is discharged from the elevator cylinder, it escapes to a discharge tank and is pumped to the accumulator, either against a confined cushion of air or against a loaded plunger, or to an elevated supply tank. The accumulator is filled automatically by a pump controlled by a governor. The water is used over and over.

Pressure on Pump Suction—Will a plunger pump work satisfactorily if it receives its water under pressure, as from another pump placed lower to help the first take water from a deep well?

P. H. W.

For a plunger pump to work satisfactorily, when there is a pressure head on its suction, its valves must have strong enough springs to close them against the pressure during the discharge stroke, and even then there may be some slipping of the pump. When one pump delivers to another, there should be some kind of an expansion chamber between them, unless both operate at exactly the same rate or there is opportunity for the initial pump to slip, as a centrifugal pump would, if the second pump failed to take the water fast enough. If the first pump does not deliver the water fast enough, the second pump must slip. The best means of equilizing is to have the first pump deliver into a standpipe to which the suction of the second pump is connected.

Boiler Heating Surface—What will be the heating surface of a Manning boiler 74 in. in diameter, with 284 two-and-one-half-inch tubes 15 ft. long and a firebox 84 in. in diameter by 55 in. high from the top of the grate to the tube sheets?

F. C. C.

Tube heating surface is usually calculated as the area corresponding to the nominal exterior diameter. The circumference of a circle of 2½ in. diameter being 7.854 in., the total heating surface in the tubes is

$$\frac{7.854}{12} \times 15 \times 284 = 2788.17 \text{ sq.ft.}$$

The firebox flue sheet being 84 in. in diameter, its gross area is

$$\frac{84 \times 84 \times 0.7854}{144} = 38.48 \text{ sq.ft.}$$

A 2½-in. tube has a cross-sectional area of 4.9087 sq.in., and the area to be deducted for the tubes is

$$\frac{4.9087}{144} \times 284 = 9.68 \text{ sq.ft.}$$

leaving as the net flue sheet area

$$38.48 - 9.68 = 28.8 \text{ sq.ft.}$$

The diameter of the furnace being 84 in., its circumference is

$$\frac{84 \times 3.1416}{12} = 21.99 \text{ ft.}$$

The height of the furnace being 55 in., the area of the sides of the firebox (neglecting the fire-door opening) is

$$21.99 \times \frac{55}{12} = 100.79 \text{ sq.ft.}$$

The total heating surface in the firebox, therefore, is

$$28.8 + 100.79 = 129.59 \text{ sq.ft.,}$$

and the total in firebox and tubes,

$$2788.17 \times 129.59 = 2917.76 \text{ sq.ft.}$$

ENGINEERS' STUDY COURSE

Mensuration—Part IV
The Circle

If you cut out a carboard disk, 1 in. in diameter, mark a point on the edge, and, placing this point downward on a sheet of paper, roll the disk until the point marked again touches the paper, you will find that the distance passed over, or the circumference, is very nearly $3\frac{1}{7}$ in., or, reduced to decimals, 3.1416 in. This number is not wholly correct, however, as it would never come out even as a decimal.*

Take a circle, 3 in. in diameter, the circumference will be 9.4248 in. The ratio will be

$$9.4248 : 3 = \frac{9.4248}{3} = 3.1416$$

This is true of any circle, so we have the rules. To find the circumference when the diameter is known:

Rule: *Multiply the diameter by 3.1416.*

To find the diameter when the circumference is known:

Rule: *Divide the circumference by 3.1416.*

Example: A flywheel 8 ft. in diameter makes 175 r.p.m. What is the rim velocity in feet per minute?

$$3.1416 \times 8 \times 175 = 4398.24 \text{ ft.} \quad \text{Ans.}$$

Example: The drivers of a locomotive are 60 in. in diameter. If they made 3200 revolutions in traveling eight miles, what was the distance lost due to slippage? 5280 ft. equals one mile.

$$5 \text{ ft.} \times 3.1416 \times 3200 = 50{,}265.6 \text{ ft.}$$

in 3200 revolutions.

$$\frac{50{,}265.6 - (5280 \times 8)}{5280} = 1.52 \text{ miles.} \quad \text{Ans.}$$

Numerous experiments have shown that each square foot of bare iron pipe surface will radiate 3 B.t.u. per hr. for each degree difference between the temperatures of the pipe and the air surrounding the pipe.

Example: What would be the loss in B.t.u. per annum, in the following lengths and sizes of pipes: 200 ft. 2 in.; 66 ft. 6 in.; 162 ft. 4 in.; 20 ft. 12 in.? Steam at 150 lb. gage (366.5 deg. F.) is in these pipes 24 hr. per day, 300 days per year. The average temperature of the air is 80 deg. F. The actual external diameters of the pipes are as follows: 2 in., 2.37 in.; 4 in., 4.5 in.; 6 in., 6.62 in.; 12 in., 12.75 in.

The writer gets 3,066,971,040 B.t.u. for an answer. Is this correct?

Areas of Circles

Let us see how the rule for finding the area of a circle is established:

Let Fig. 1 represent a circular piece of cardboard, the lower half ABC cut into eight equal pieces or sectors, and spread out, as shown at Fig. 2. It is obvious that the length ABC, Fig. 2, equals ABC, Fig. 1, or half the circumference of the cardboard and the height or altitude of the sectors will equal the radius of the circle. If the upper half of the circle be cut into eight equal sectors, they will fit into the spaces formed by the sectors from the lower half, and we will have a nearly perfect rectangle, the area of which would be the same as that of the circle. The greater the number of sectors, the more perfect would be the rectangle formed.

As the length of the rectangle formed will equal one-half the circumference, and the width will equal the radius of the circle, we have the following rule for finding the area of a circle.

Rule: *Multiply one-half the circumference by the radius.*

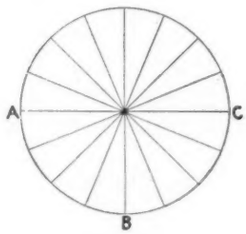


Fig. 1

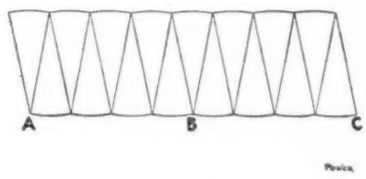


Fig. 2

Expressed as an equation,

$$Area = \frac{3.1416 \times D}{2} \times r$$

but, since the diameter is twice the radius, or $\frac{D}{2} = r$

$$Area = 3.1416\ r^2$$

So the rule might also be given.

Rule: *Square the radius and multiply by 3.1416.*

or

Rule: *Square the diameter and multiply by 0.7854.*

The figure 0.7854 is obtained by dividing 3.1416 by 4. The reason for dividing by 4 is evident from the last equation, where $\frac{D}{2}$ must be substituted for r if the area is to be determined from the diameter instead of the radius. Making the substitution

$$Area = 3.1416 \left(\frac{D}{2}\right)^2 = \frac{3.1416 \times D^2}{4} = 0.7854\ D^2$$

A graphical explanation may be given based on the fact that the line which bounds a square is to the line which bounds a circle of the same diameter as the square, as the area of the square is to the area of the circle.

Let Fig. 3 represent a square 1 sq.in. in area. If we draw a circle that will just fit inside the edges of this square, as in Fig. 4, the diameter will be 1 in.

We know the length of the line which bounds the square is 4 in., and, from what we have learned of 3.1416, or π, we know that the line which bounds the circle is 3.1416 in.

*Some one has taken the trouble to carry it out to 127 places: 3.14159265358979323846264338327950288419716939937510582097494459230781640628620899862803482534211706798214808651327230664709384 46+. In 1873, a Mr. Shanks carried it out to 707 places.

Now, expressing all this as a proportion we have

$$4:1 = 3.1416:X$$

or the area of the circle.

Dividing the product of the means to find the other extreme we have

$$\frac{1 \times 3.1416}{4} = 0.7854 \; sq.in$$

We have now proved that the constant 0.7854 represents the area in square inches of a circle 1 in. in diameter. If the area of the circle is to be figured in square feet, square yards, square miles, etc., this constant is also used because the relative value remains the same no matter what unit of measure we may use.

FIG. 3

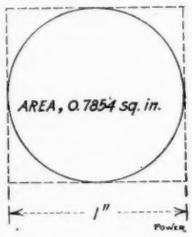


FIG. 4

RINGS

The ring, Fig. 5, is really a cylinder bent into a circular form, so the volume will be the same as that for a cylinder whose altitude is equal to the mean circumference or the length of the dotted line A. The mean circumference will be twice the distance OA multiplied by 3.1416 or π.

If the radius or distance $OA = R$, and the radius of the cross-section $BA = r$, the formula for the volume of the ring will be

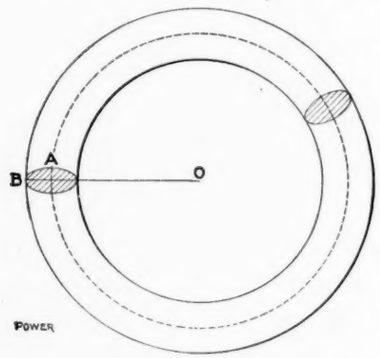


FIG. 5

$$V = \pi r^2 \times 2 \pi R$$

and for the area of the ring's surface

$$A = 2 \pi R \times 2 \pi r$$

Fig. 6 shows a flat, circular ring like a gasket.

It is easy to see that to find the area of one face we proceed as follows:

Rule: *From the area of the large circle subtract the area of the small circle.*

The area of the large circle will be

$$12^2 \times 0.7854 = 113.09 \; sq.in.$$

The area of the small circle will be

$$6^2 \times 0.7854 = 28.27 \; sq.in.$$

and the area of the flat ring will be

$$113.09 - 28.27 = 84.82 \; sq.in.$$

A ring of rectangular section is shown in Fig. 7. The volume of such a ring may be found similarly as for the circular ring, being equal to the volume of a prism whose altitude equals the mean circumference of the ring. This is so because this ring is really a prism bent into a circle.

If

R = Outer radius;
r = Inner radius;
T = Breadth of face;

we have the following formulas for:

$$Mean\ radius = \frac{R + r}{2}$$

$$Mean\ circumference = \pi\ (R + r)$$
$$Area\ of\ cross\text{-}section = (R - r)\ T$$
$$Volume = (R - r)\ T\ \pi\ (R + r)$$

Superficial area or area of all four sides =

$$\pi\ (R + r) \times 2\ [(R - r) + T]$$

Another, and perhaps the commoner, method of finding the volume or area of a square ring, is to find the area of one face, as was done for the gasket or flat ring, and multiply by the thickness. The volume would then be equal to

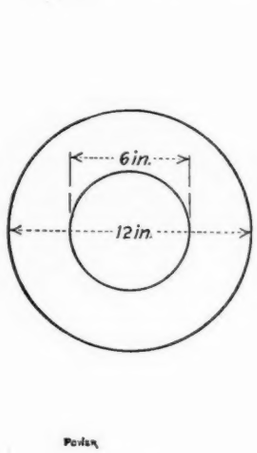


FIG. 6

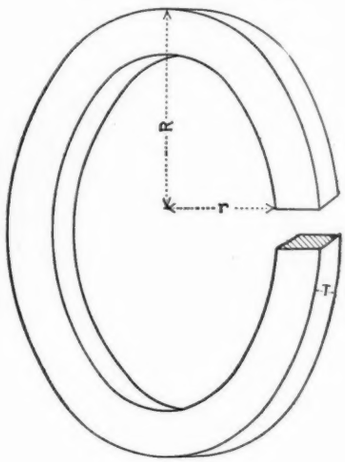


FIG. 7

$$(\pi\ R^2 - \pi\ r^2)\ T$$

and the total or superficial area to

$$2\ (\pi\ R^2 - \pi\ r^2) + T\ (2\ \pi\ R) + T\ (2\ \pi\ r)$$

Example: A flywheel rim has an outer radius of 144 in., an inner radius of 139 in., the breadth of the face is 20 in. What will the rim weigh, cast iron weighing 0.26 lb. per cu.in.?

We see that the formula for volume must be used.

$$Volume = (R - r)\ T\ \pi\ (R + r)$$

$R = 144$ in.;
$r = 139$ in.;
$T = 20$ in.;

substituting these values, we have

$$Volume = (144 - 139)\ 20 \times 3.1416 \times (144 + 139)$$
$$= 88{,}907.28 \; cu.in.$$

The weight will be

$88{,}907.28 \times 0.26 = 23{,}115.89$ *lb.* Ans.

Example: What will be the area of all four sides of this flywheel rim?

The formula is

$$Area\ \pi\ (R + r) \times 2\ [(R - r) + T]$$

substituting the known dimensions we have

$$Area = 3.1416\ (144 + 139) \times 2\ [(144 - 139) + 20]$$
$$= 44{,}453.5\ sq.in.$$

Check for correctness the formulas used in the last two examples, by working the same problems with the other formulas given.

Examples for Practice

(1) A winding engine has a drum 8 ft. in diameter. In what time will the engine haul a cage up a 482.6-ft. shaft if the drum makes 20 r.p.m.?

(2) How many square inches will there be in a gasket for the flanges of a 6-in. pipe, area occupied by the bolts to be deducted? (A flange for a 6-in. pipe is 11 in. diameter and has eight ⅞-in. bolt-holes.)

(3) What will be the weight of a cast-iron pipe 12 ft. long, 2 ft. in outer diameter and 1 in. thick; cast iron weighing 0.26 lb. per cu.in.?

Answers to Last Week's Problems

(1) (a) 237.945.6 gal.
(b) 1 hr. 9 min. 4 sec.

(2) 476,367 cu.in.
(a) 2062 gal.
(b) 17,176.5 lb.

Where the Sediment Forms

The sediment in a boiler will accumulate in the portion of the bottom near to the region of the bridge-wall in the furnace; and if you have no such thing as a bridge-wall it will accumulate where the fire is hottest. This is a result of heat movements that send the hot water upward from highly heated spots while the cooler water surrounding sweeps in below, carrying with it sediment that builds a little mound; this, of course, if there is some one spot that is materially hotter than the rest of the surface exposed to the fire.

We all know the result; that the sediment forming in a heap keeps the water from the iron at this very spot till the heat causes the iron to bulge down into what we call a bag; but there is a lesson as well as a result. In the first place, the fireman should strive to have his heat distributed as near equally over the bottom of the boiler as is possible. It is not practical to get it so, strictly speaking, but a careful man can so manage his furnace and fires that the heat will be so well distributed that there will not be much of this accumulating of sediment in one spot, and then he can keep his boilers a little cleaner; and that will help some, too.—*National Engineer.*

Safety suggestions from employees are invited by the safety committee of the Lehigh Valley Railroad. The employee who makes a suggestion, whether or not it is adopted, receives a report as to what was done about it, with the reasons therefor. On each division the man who makes the best suggestion in the year toward increased safety of operation is given an "honor button" and a month's vacation with pay. In addition annual passes will be granted men with clear safety records for a year.

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

R. O. Richards charges us with "dangling." Then he dangles this dinged dangle before us:

"Editor of the jays (J. J. J. J.) column:

"In the Feb. 18 issue, I note that you dangle before the eyes of your readers two ways of making easy (?) money. I thank you. In return, let me tip you to a far easier way. Uncle Sam is about to issue a new five-cent piece. Collect 1913 nickels; they are worth $95.65.

"Framingham, Mass. R. O. RICHARDS."

R. O., your initials stand for "Ring Off"! You've got the wrong number if you think we're not hep that

1913 × 0.05 = 95.65 (perfectly good dollars).

It almost surpasses belief that a lumber-mill engineer should possess one million real green dollars, but that's what the papers say of James Bennett, of Marshfield, Ore. Of course, this sum does not represent his savings; it was willed to him. They also say, "James does not wish to leave his job." Stick, Jimmy, stick! Buy the sawmill and you will have something handy when you start cutting coupons.

"THE CONDITIONS IN MASSACHUSETTS"

EXAMINING BOARD
Determine the rotative effort of the tangential component of the angular thrust of the connecting rod on the crank pin
That'll stick 'em all right
IVIL SERVICE AMINATION
Gee Bill, they've got stuff in this examination I never heard of before!

"THE EXAMINERS EXAMINED"

Once in a while, someone gets even with the dentist. A lady named Peck got a bushel of money ($384) in a suit (a "law" suit) from Dr. Stevens because he pulled the second upper left bicuspid when he should have pulled—some other. Steve Brodie took a chance and so did the other Steve, but there's more than one kind of bridge work and more than one kind of Steve gets gotten. Get it?

In this column, on Feb. 18, we asked our readers to "dig up a postcard and name offhand the names of ten men famous in the history of steam engineering. Prof. William Kent promptly responds with this list.

Savery—First practical pumping engine.
Newcomen—Cylinder and piston reciprocating engine.
Watt—Separate condenser, rotative engine, governor.
Oliver Evans—High-pressure, non-condensing engine.
Fulton—Application of Watt's engine to steamboat.
Stephenson—Improvements in locomotive.
Sickels—Drop cutoff.
Corliss—Attaching drop cutoff to governor.
Porter—High-speed engine.
De Laval—First commercial steam turbine.

Prof. Kent adds Parsons and Stumpf, for good measure.

Oil as a Fuel

By Reginald Trautschold

SYNOPSIS—Discussion of the use of oil as a fuel for generating steam. Emphasizes its advantages over coal in greater convenience and cleanliness and less labor of handling, and even claims equal or better economy in many sections not at present regarded as favorably located to burn oil.

About half a billion tons of coal is now mined in the United States per year, including bituminous and anthracite coals of all grades. Of this amount probably two-fifths is used to generate steam for use in stationary engines. At the same time, the yearly production of crude oil is close to three hundred million gallons, of which a considerable part is being used to develop power in three ways: By using the crude oil itself in internal-combustion engines of the Diesel type; through the explosive use of certain byproducts, as in gas and gasoline engines; and as fuel for generating steam for use in steam motors. The last of these three will be discussed in this article.

This use of oil as fuel does not appear to be as universal as it might be, probably due to lack of general knowledge of the efficiency that can be expected when such fuel is carefully and scientifically used and to the fact that fuel oil is not obtainable at a reasonable price at all times. The first of these drawbacks should be easily surmountable and it will be the object of this article to present some general information and facts pertaining to the question. The second drawback is more difficult to overcome but, as it can be controlled to a certain extent, should not bear sufficient weight to unreasonably retard the more extensive use of this exceedingly convenient and usually economical fuel.

Substituting fuel oil for coal in the production of steam, evades many of the drawbacks of adopting other means of generating power in the ordinary existing steam plant. No unheaval in the engine room, accompanying discard of valuable equipment, addition of new engines, etc., nor radical changes in the engine-room force and management are necessary. This does not mean that the field lost to the gas engine, where natural gas or gases from oil refineries are procurable, can be regained by using oil as a fuel, nor that the surprising economy of certain internal-combustion engines of the Diesel type can be equaled by any possible gain in economy in steam generation, due to using oil fuel. It does mean, however, that oil fuel must now be considered as a serious competitor of coal in many localities where the power plant depends for its operation upon steam.

Before considering the requirements for an economical oil-burning system it will be well to clearly understand

What Constitutes a Good Fuel Oil

or what its specifications should be. This will anticipate the common criticism that fuel oil is an unsafe fuel, subject to accidental ignition, etc. Fuel oil is frequently referred to as crude oil and possibly this is the reason for the erroneous idea of its danger, for crude oil is a fuel oil only when its flashing point is high and the oil comparatively free from gasoline and naphtha. Crude oil itself is a mixture of a number of hydrocarbons which may be separated through the agency of heat and in a crude oil with *low* flashing point the first distillate gases given off in refining are very explosive so that such an oil cannot be safely used as a fuel oil.

Gasoline is driven off at a temperature of from 140 to 160 deg. F., during the distillation of crude oil, and at a temperature of 200 deg. F. the lighter benzines or naphthas have been separated from the crude. The residue, should distillation be stopped at this point, would be an oil that could be handled as safely as coal, but the specifications for fuel oil generally call for a fire test of 250 to 300 deg. F., such oil contains little naphtha and cannot ignite accidentally if ordinary precautions are taken. A shovelful of red-hot ashes can be thrown into a tank of such fuel oil without danger, or a red-hot poker may be used to stir it without fear of an explosion or ignition.

Following in the order in which the various oils are distilled from crude comes kerosene, which constitutes about one-half the mixture. This is also a good oil for fuel but is naturally considerably more expensive than fuel oil and cannot be, therefore, considered as a practical substitute for coal for any but very small engines. The present discussion will be limited, therefore, to a brief description of the equipment and requirements for a fuel-oil system.

Equipment for Oil Burning

Between a steam plant operated on coal and one on oil fuel, the only difference lies in the boiler rooms; the engine rooms are identical. The alterations or additions in the boiler room are neither radical nor complicated so that a plant operating on coal can easily, quickly and inexpensively be converted into an oil-burning plant by adding a few comparatively simple mechanisms and equipment. All that is required is the installation of oil burners in the furnace doors; proper protection of the grates and all exposed metal parts of the firebox with firebrick; a reservoir or storage tank for confining the oil; apparatus for pumping the oil to the burners, with means of regulating the pressure at which the oil is delivered and facilities for preheating the oil before delivery; a steam connection between the boiler and oil burner for vaporizing the oil; and the piping that may be necessary for connecting up the system.

Without going into an extended analysis of the thermal value of fuel oil or of ways of firing such fuel, it may be stated that one gallon of good fuel oil per hour will furnish sufficient heat to develop from 2½ to 3 hp. Oil burners, proportioned to so regulate the consumption of oil, are procurable in a compact form with connections for the supply of oil under suitable pressure and for the supply of steam to vaporize the oil as delivered from the burner. Simple hand valves, forming part of the mechanism of the burners, regulate the supply of both steam and oil, the first of which should be delivered under constant pressure for hand regulation of the furnace and at variable pressure if mechanical regulation is used.

To facilitate atomizing the oil and to maintain the constancy of its supply under given pressure, it should

be preheated to reduce its viscosity and thus the pressure required to assure proper delivery. The steam required for vaporizing is obtained directly from the boiler, when in operation, and its pressure should be reduced to an equivalent of from 3½ to 4½ times that at which the oil is supplied to the burners. The apparatus for preheating the oil, subjecting it to pressure, etc., is usually in the form of one compact unit; has considerable capacity; is comparatively inexpensive and is all the mechanism that is required to complete the system. The oil-storage tank preferably of a capacity sufficient to take care of the oil requirements for a considerable period, if the price of ful oil fluctuates to any considerable extent or if oil is not always readily obtainable in the locality of the power house, is also comparatively inexpensive and should be

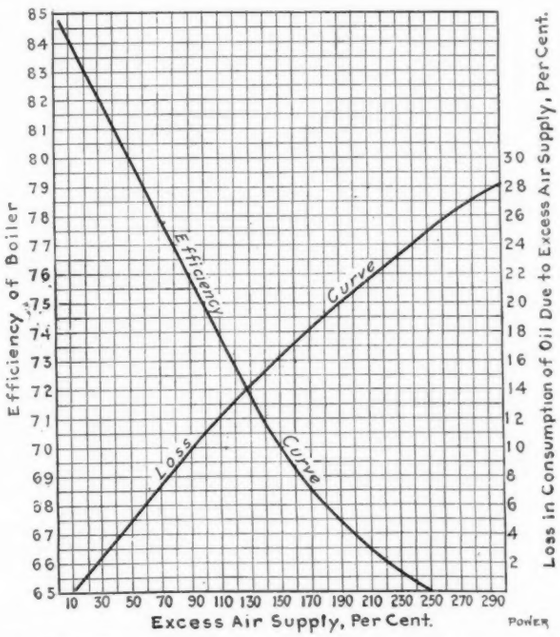

Efficiency of Boiler
Loss in Consumption of Oil Due to Excess Air Supply, Per Cent.
Excess Air Supply, Per Cent.
Efficiency Curve
Loss Curve
POWER

CHART I. EFFECT OF AIR SUPPLY ON EFFICIENCY OF BOILER AND CONSUMPTION OF OIL—OIL-FIRED BOILERS

installed underground, with proper vents, etc., so as to minimize the insurance burden. The piping connecting up the various parts of the system should also be installed underground when possible and made as simple as the installation will allow.

THE EFFICIENCY OF THE SYSTEM

depends, to a great extent, upon the supply of air for combustion. This is important no matter what kind of fuel is burned, but is especially so with oil fuel, as upon it largely depends the economic value of the installation. The boiler efficiency of the average plant burning coal is much more frequently around 40 or 50 per cent. than 60 or 70 per cent., and only the exceptionally efficient installation will show over 70 per cent. With oil, this boiler efficiency can be greatly increased, owing to the possibility of concentrating the heat and the decreased loss due to radiation. A boiler radiation loss of but 3 per cent. or better can be obtained, in an oil-fired boiler, by proper conservation of heat in the fire chamber by covering all metal parts, not in direct contact with the water, with good quality firebrick and controlling the supply of air in the combustion chamber.

A given supply of oil requires a specific volume of air for complete combustion (theoretical air supply) but practically the best efficiency is obtained with an excess air supply of about 10 per cent. With this excess a boiler efficiency of about 85 per cent. may be realized in an oil-fired boiler with properly proportioned firebox carefully protected with good quality of firebrick and the flame of the burner directed against a correctly located wall of firebrick and so as not to touch the metal of the boiler. An air supply of 10 per cent. over the theoretical requirement, however, is a relatively small volume and several fold this amount may easily be admitted unless the disadvantage is fully realized by the operator or furnace man, and constant care taken in regulation. An excess air supply of only 300 per cent. will reduce the boiler efficiency to under 65 per cent., or about that which would ordinarily be obtainable with coal as fuel, and would necessitate burning nearly 30 per cent. more oil to maintain the requisite boiler pressure—see Chart I. Here evidently lies the key to the economy of the oil-burning system, for one properly installed and operated will develop the same power from a boiler capacity of 60 to 80 per cent. of that required when burning coal.

EFFICIENT OIL BURNING

is indicated by the appearance of the combustion chamber and the condition of the exhaust passing to the smoke-

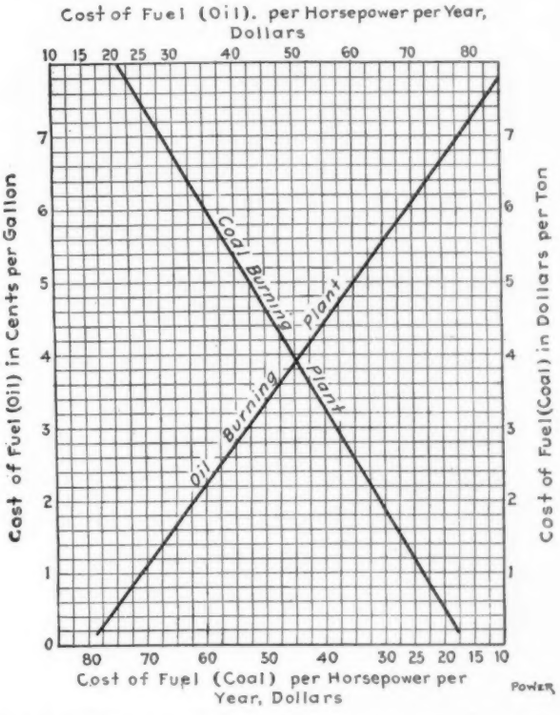

Cost of Fuel (Oil). per Horsepower per Year, Dollars
Cost of Fuel (Oil) in Cents per Gallon
Cost of Fuel (Coal) in Dollars per Ton
Cost of Fuel (Coal) per Horsepower per Year, Dollars
Coal Burning Plant
Oil Burning Plant
POWER

CHART II. AVERAGE COST OF STEAM POWER, BASED ON A PLANT OF 500 HP.

stack. The combustion chamber should show a clear white incandescence, with only a slight apparent flame, and no smoke or unburned gases should pass to the smoke-stack. Such conditions can only be recognized through practice, but, when the correct appearance is known to the furnace man, they can be quite easily maintained.

The economy in oil consumption is then dependent upon the minimum supply of oil requisite to maintain the desired pressure on the boiler. This, in an installation where the load is fairly constant, can be controlled by

hand by an experienced man, and then the supply of oil is regulated under constant pressure and the air supply by hand-operated dampers. Where the load fluctuates considerably, hand regulation is difficult and mechanical control should be installed. In a mechanically controlled system the oil outlet in the burner is maintained constant and the supply of oil regulated by the delivery pressure. This variation in oil pressure, which is directly dependent upon the fluctuations in the load, is made use of to regulate the admission of air to the combustion chamber, an increase in load producing an increase in oil pressure and a corresponding enlargement of the air-admission passage and *vice versa*.

TABLE I. AVERAGE FIXED CHARGES OF THE POWER HOUSE

Based on Plant of 500 Hp.

Oil Burning Plant

Engine room:—		
Building, etc	$10	
Engines, accessories, piping, etc	30	
Foundations, installation, etc	5	
		$45 per horsepower
Depreciation 5% total cost		
Repairs 2%		
Interest 6%		Total, per horsepower per year.
Insurance 1%		Item A—$7.20
Taxes 2% ¾ cost		
Boiler room:—		
Building foundations, etc	$4.50	
Chimneys, flues, etc	7.00	
Boilers, etc	7.50	
Oil burning systems (complete)	3.00	
		$22 per horsepower
Depreciation 5% total cost		
Repairs 2%		
Interest 6%		Total, per horsepower per year,
Insurance 2%		Item B—3.63
Taxes 2% ¾ cost		
Cost of operation:—		
Engine room,		
Attendance	$1.80 per horsepower per year.	
Supplies	0.80	
	2.60	Item C—2.60
Boiler room:		
Attendance	$1.10 per horsepower per year	
Supplies	0.47	
	$1.57	Item D—1.57
Total fixed charges, per horsepower per year		$15.00

Coal Burning Plant

Engine room:—		Item A—$7.20
Boiler room:—		
Building, foundations, etc	$5.00	
Chimneys, flues, etc	8.00	
Boilers, feed pumps, etc	12.00	
		$25 per horsepower
Depreciation 5% total cost		
Repairs 2%		
Interest 6%		Total, per horsepower per year,
Insurance 1%		Item B—3.87
Taxes 2% ¾ cost		
Cost of operation:—		
Engine room:		Item C—2.60
Boiler room:		
Attendance	$1.90 per horsepower per year	
Supplies	0.90	
	$2.80	Item D—2.80
Total fixed charges, per horsepower per year		$16.47

The mechanical operations of such regulators differ, but usually the apparatus consists of a system of levers operating the damper rocker shaft and controlled by a diaphragm actuated by pressure in the oil connection to the burner. Using the variation in oil pressure to regulate the air supply gives a sensitive and reliable control that when once adjusted for the range of a particular installation will insure efficient and economical oil consumption.

An efficient oil-burning system for boilers compares favorably with the ordinary coal burning from an economic standpoint as is indicated in Chart II. This chart is plotted from Tables I, II and III, which were compiled from various data pertaining to steam plants of about 500-hp. capacity and are sufficiently accurate to permit a comprehensive comparison between plants of that size operating on oil and on coal.

Chart II also affords means for forming a conservative opinion as to the relative economies of any size plant operating on the two fuels with the respective fuels at various prices, for the relation existing between the cost of fuel and the average cost of power is practically constant in any ordinary size of power plant under efficient management, since in larger plants the fixed charge per unit of power is reduced and the consumption of steam is relatively less while in smaller plants the reverse is true.

TABLE II. AVERAGE COST OF FUEL FOR THE POWER HOUSE

Based on Plant of 500 Hp.

Fuel Oil		Coal	
Per gal.	Per hp. per year	Per ton	Per hp. per year
$0.01	$9.00	$1.00	$7.20
0.015	13.50	1.50	10.80
0.02	18.00	2.00	14.40
0.025	22.50	2.50	18.00
0.03	27.00	3.00	21.60
0.035	31.50	3.50	25.20
0.04	36.00	4.00	28.80
0.045	40.50	4.50	32.40
0.05	45.00	5.00	36.00
0.055	49.50	5.50	39.60
0.06	54.00	6.00	43.20

TABLE III. AVERAGE COST OF STEAM POWER

Based on Plant of 500 Hp.

Oil Burning Plant		Coal Burning Plant	
Cost of Oil per gal.	Steam Power per hp. per year	Cost of Coal per ton	Steam Power per hp. per year
$0.01	$24.00	$1.00	$23.67
0.015	28.50	1.50	27.27
0.02	33.00	2.00	30.87
0.025	37.50	2.50	34.47
0.03	42.00	3.00	38.07
0.035	46.50	3.50	41.67
0.04	51.00	4.00	45.27
0.045	55.50	4.50	48.87
0.05	60.00	5.00	52.47
0.055	64.50	5.50	56.07
0.06	69.00	6.00	59.67

Advantages of Oil Fuel over Coal

Oil is more easily and cheaply handled than coal, can be stored at less expense and when stored does not deteriorate, minimizes standby losses, and is more convenient and cleanly. Of even greater economic importance are decrease of loss of heat generated in the furnace, due to concentration and more even distribution of heat, and diminished radiation losses; the fact that the boiler and its tubes are kept free from nonconducting deposits, such as are precipitated from the gases in coal burning; the conservation of heat lost when burning coal because of the necessity of opening the furnace door to fire; the ease with which the intensity of the fire may be regulated; and the rapidity with which steam may be raised. Still other benefits are the saving in labor and the absence of expense entailed in handling and disposing of ashes. These advantages are represented in the increase of 20 to 40 per cent. in the steaming capacity of the boiler fired on oil, to which reference has already been made.

The possible economy in fuel consumption from the use of fuel oil and the numerous benefits accruing from its use would doubtless bring about its rapid introduction but for the

Unreliability of Supply at Constant Price

This condition does not exist in sections of the country near oil wells and there the use of oil has already become general. In many other parts of the country the average price of fuel oil compares favorably with that of coal, but the fluctuations in the price of oil are great. Even here a judicious purchase of oil when cheap and maintaining a comfortable reserve for use during periods of elevated

prices will enable relying on oil for fuel without paying an exhorbitant price for it, for at some time during each year oil is usually procurable at a favorable price for economic power development.

One influence at work tending toward a more uniform price for fuel oil is the steadily increasing demand for gasoline brought about by the automobile. An excellent fuel oil is left as a residue after distilling gasoline from crude oil and a constant demand at a uniform price for gasoline will mean a constant and low price for fuel oil unless the demand for other heavy byproducts of crude oil should unexpectedly increase.

❋ ❋

Signification of Torque

By Franklin Van Winkle

SYNOPSIS—Terms expressive of rotary power may at first be misty to engineers whose experience has been confined to reciprocating engines. Ambiguity in use of the term torque has led most Continental authors to discard the word from their writings. The article describes the method of measuring the torque of a motor and defines the relation of torque to horsepower and kilowatt.

❋

Many engineers upon first acquaintance with electrical machinery reserve their confidence in ordinary switchboard indicators as means for determining power output. It requires, however, only short practical experience in care of electrical generators and motors to convince the most skeptical that the voltmeter, ammeter and wattmeter are as faithful monitors of power product as the steam-engine indicator is of the performance of the reciprocating engine.

"Kilowatt" soon becomes as familiar an expression for amount of power as "horsepower." But after long practice in computing power developed by the piston of a reciprocating engine, one is apt to hold in mind, as a

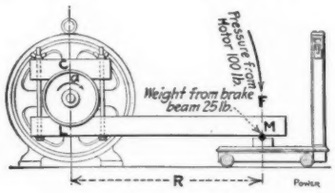


Fig. 1.

fundamental concept, that power consists of work performed in a straight line, and engineers who have been reared in the gospel of the old school of steam engineering are prone to consider that the electric-motor and steam-turbine practice have brought into the power field methods of computing power that are not entirely reconcilable with the older methods of computation. It is erroneous to assume, however, that the era of rotary motors has brought with it any actual innovations in dynamical computations, for it is rather from motives of greater convenience than novelty that the terms expressive of energy of rotary power are introduced. For most part they are only modifications of expressions which are quite as ancient to the mathematician and physicist as the time-honored P-L-A-N formula is to the steam engineer; anyone versed in the old power nomenclature who will give a little attention to the subject of motors, soon finds himself using terms and formulas for estimating rotary power with as much facility as he ever had employed data obtained with the steam-engine indicator.

Probably no term used in connection with generators and motors has given rise to more confusion in the minds of operating engineers than "torque," and this is probably because the word is used with two analogous though different meanings, viz., one signifying the mechanical rotary or turning force which acts on the armature of an electric generator or motor causing it to rotate, or in general, the forces employed in rotating the armature; and another which signifies the summation of the moments or leverages of the turning forces applied to an electric generator or other machine which turn it or cause its rotation.

The term torque is, therefore, used to stand for both (a) a mechanical "force" or "pull" at the periphery of the armature, such as is known in mechanics as "tangential effort"; and (b) the moment or leverage of the tangential effort, i.e., the tangential effort multiplied by its radius of action. This ambiguity has caused most Continental writers to discard the word "torque" altogether. But with us the word is so generally used in both senses that the American engineer must differentiate the significance of the term as he may chance to meet it.

In referring to details of construction of generators or motors, one may have occasion to use the word in its narrower sense of "tangential effort," but as a power-plant engineer he is concerned with the broader sense of the term, viz., as a moment of tangential effort. Employed in the latter sense, the torque of a generator or motor usually signifies equivalent pounds of pull at the end of a radius or arm 1 ft. long.

Most operating engineers know the meaning of brake power and if they will trace out the relations which these definitions bear to brake power, the term torque will be regarded as a novelty only in name.

Suppose, as indicated in Fig. 1, a motor has a driving pulley which rotates in the direction of the arrow *a*. A clamp brake *CLM*, suitable for resisting rotation of the

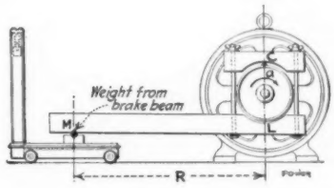


Fig. 2.

driving pulley, affords means for determining torque and for testing the power developed by the motor when it takes a given current, if account is taken of (1) the horizontal distance, *R* (as 4 ft.) from the center of the driving pulley to the point *M*, where the brake arm is supported on a platform scale, (2) the force *F* (as 100 lb. pressure) with

which the action of the motor presses the brake arm down on the scale, in addition to that weight of the brake beam which is supported on the scale, and (3) the number of revolutions N (as 500 per minute) made by the motor in a stated time.

If now the brake beam is reversed and when freely

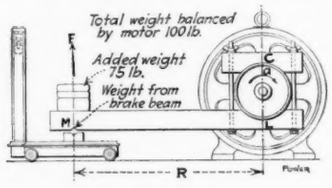


FIG. 3.

supported with the motor at rest as shown in Fig. 2 the pressure due to the brake beam alone is observed, and weights T are added to the arm, as shown, Fig. 3, so as to make the total weight the same as the downward pressure which was created on the scale in addition to the dead weight of the brake arm when the motor was driven as shown in Fig. 1, then the brake would be suitably loaded for creating the same resistance as the resistance overcome in Fig. 1. Then if the motor were supplied with the same current as before, the clamp of the brake could be tightened so as to just hold the brake arm poised as shown in Fig. 3.

In either case it is to be recognized that the rotary effort exerted by the motor would be precisely the same as though the motor pulley consisted of a drum, as shown in Fig. 4, having a radius R equal to the effective length R of brake arm as in Figs. 1, 2 and 3, and the effort would be the same as if the motor were actually employed in

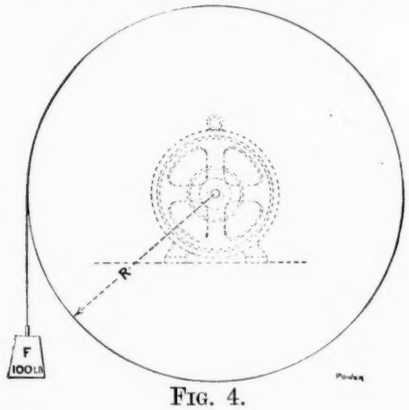


FIG. 4.

raising a weight F of 100 lb. through a cord or cable suspended over the periphery of the drum.

If the distance R is 4 ft. then for each revolution of the drum the weight would be raised through a height equal to the circumference of the drum, i.e., the distance would be $2\pi \times R$ or

$$(2 \times 3.1416) \times 4 \text{ ft.} = 25.1328 \text{ ft.}$$

and if the weight amounted to 100 lb., then the motor would develop.

$$2\pi \times (R \times W) = 2 \times 3.1416 \times (4 \times 100) = 2513.28 \text{ ft.-lb.}$$

for one revolution.

The quantity $R \times F$ is called the moment of rotation and to raise the weight, as in Fig. 4, or overcome an equivalent brake resistance, as illustrated in Figs. 1 and 3, it is evident that an equal and opposite moment must be exerted tangentially by the motor in the direction of the arrow a, Figs. 1 and 3. The pull thus required will depend on its distance of action from the center of the motor shaft, for the tangential force multiplied by its distance would have to equal $R \times F$.

It is evident that in considering the power of a motor, the term torque should be used only as meaning the product of $R \times F$, as far as concerns exertion of turning effort, it is inexplicit to use the term to signify pull at the periphery of the armature unless the radius of the armature is distinctly understood.

In any case the same turning effort would require $R \times F$ to have the same value and to avoid ambiguity, torque should be restricted to mean *turning moment*.

If the armature has a radius of R ft. and a pull at the periphery of F pounds in any case

$$2\pi \times (R \times F);$$

then if we know the pull at the circumference of the armature and the radius of the armature the torque will be the product of these two factors.

In each of the cases illustrated by Figs. 1, 3 and 4 it is to be seen that the energy exerted per revolution was

$$2\pi \ (R \times F);$$

i.e., the number of foot-pounds exerted per revolution would be $2\pi \times$ the torque, and consequently $2\pi \times$ torque $\times$ r.p.m. = ft.-lb. per min.; or, taking the figures quoted, with 500 r.p.m. the number of foot-pounds developed per minute would be

$(2 \times 3.1416) \times (4 \times 100) \times 500 = 1{,}256{,}640$ *ft.-lb.* Dividing this amount by 33,000 we obtain 38.08 hp.; or dividing by 44.24 we obtain 28,405 watts, or 28.4 kw.

From the above it appears that when torque is regarded as the moment of rotation, then

$$\frac{2\pi \times torque \times r.p.m.}{33{,}000} = horsepower$$

and

$$\frac{2\pi \times torque \times r.p.m.}{44{,}240} = kilowatts$$

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To Encourage Cleanliness in Plants

A good way, if not the best way, to encourage cleanliness of machines and of the factory in and about the machines, is to keep conveniently at hand the implements necessary to secure cleanliness—brooms, brushes, cotton waste, etc. In some plants the sweeping is given over to some one man on each floor, and where this is done it may not be advisable to provide a broom convenient to every machine man. But one thing that should always be handy and within reach is cotton waste, something to clean the oil and dirt off of parts of machinery. This should be kept where it is easily accessible.

It is better to have them destroy and waste a little of it unnecessarily than it is to be stingy with it and let the machines suffer for want of cleaning. Both waste and receptacle for the waste should be provided—that is, a box, or preferably a metal can, for holding the clean waste, and also a metal receptacle for the dirty waste. This oily waste is a dangerous thing to have thrown around carelessly in the corners, and should always be kept in a covered metal box or can of some kind to prevent danger from fire. It also contributes to neatness and orderliness.—*Wood-Worker.*

Accident to Twin Tandem Engine

A peculiar accident happened to a twin-tandem-compound Corliss condensing engine running at 78 r.p.m. and developing about 1800 hp., in the plant of the Berkshire Cotton Mfg. Co., Adams, Mass., on Feb. 11. According to our correspondent, one of the cross-head shoes came loose in some manner, broke in two pieces—one piece flying out through the engine-room door and the other barely missing the head of Howard Kaiser, assistant engineer. The mill was shut down a few hours while a new shoe was fitted.

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Piston Explodes

Another instance of where "they did not know it was loaded" is related by a correspondent. In a machine shop at Hadley, Alaska, on Jan. 22, two men were heating a piston and piston rod in a forge, preparatory to straightening the rod, when the piston exploded. Fortunately, no one was injured and the damage done was slight. The explosion is said to have been due to the presence of water in the piston. On the application of heat, the pressure due to the formation of steam caused the explosion.

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Blowoff Tank Explodes

A rather odd accident happened at the Colorado Springs high-school building, on Jan. 30. The heating plant has two 14-ft. by 60-in. boilers, operated at 60 lb. steam pressure so as to drive an engine and fan. The blowoffs from these boilers are connected to a small cast-iron reservoir 3 ft. in diameter and 3 ft. deep, which in turn is connected to the city sewer by means of a 4-in. siphon reduced to 2 in. just outside the reservoir. The 2-in. pipe has one elbow before it reaches a 4-in. sewer pipe 4 ft. away.

A week previous to the accident, the blowoff pipe on one of the boilers had become clogged between the boiler and reservoir. The engineer could not shut down and take out the pipe because of the very cold weather at that time, so he tried forcing it out by opening the blowoff under 60 lb. pressure. As usual, the space between the back of the boiler and the partition wall was narrow and the reservoir was situated just at the entrance of this narrow alley.

As the engineer opened the mud valve he heard the rush of water as the pipe cleared itself and he was just on the point of backing out and letting it blow for awhile when there was an explosion, filling the boiler room with hot water and steam, and before he could get the valve closed he was badly scalded.

The pipe from the reservoir to the sewer had become choked so that the pressure increased enough in the reservoir to rupture the ½-in. cast-iron cover. Two pieces hit the ceiling with force enough to go through, but were stopped by the floor above.

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Our juvenile contemporary "Blow-Off" had an article recently on "Taking Care of Little Things" which at first sight we thought bore on the wage question, but it didn't. Little "Blow-Off" is hitting right from the shoulder on other matters. Bore in, neighbor, you've got lots of steam behind your punch.

NEW PUBLICATIONS

THE INDICATOR HANDBOOK. A practical manual for engineers by Chas. N. Pickworth, Whitworth scholar, editor of the "Mechanical Works," etc. Part II. The indicator diagram, its analysis and calculation. Pages, 139, 5x7, illustrated, cloth, price 3s. Emmott & Co., Ltd., Manchester, England. D. Van Nostrand Co., New York.

The Pickworth handbook appears to be the favorite English manual of the steam-engine indicator. The resetting for the fifth edition has furnished the author an opportunity to extend the work in several particulars, especially the chapter dealing with diagrams from gas and oil engines. He points out that in the first edition of his work a gas-engine diagram was reduced to logarithmic coördinates, and the value of n for the compression curve determined, as has just been done by Mr. Clayton.

MATERIALS AND CONSTRUCTION—A Textbook of Elementary Structural Design. By James A. Pratt. P. Blakiston's Son & Co., Philadelphia, Penn. Cloth; 196 pages, 5x7 in.; 85 illustrations; tables. Price, 90c.

The book gives an elementary treatment of materials and the laws of construction. It is intended for trade-school students so that no attempt has been made to cover any advanced engineering work. The following list of chapters will give some idea of the contents: Elementary Principles, Materials, Elementary Calculations and Properties, Beam Design, Columns, Torsion, Action of Elementary Forces and Their Consideration in Design, Proportions of Knees and Counters, Riveted Joints, Reinforced Concrete, Tables and Data. In correctness and simplicity of treatment the book fulfills its purpose, and should be of value to the student interested in mechanics.

SHOP MATHEMATICS—Part I: Shop Arithmetic Prepared in the Extension Division of the University of Wisconsin. By Earle B. Norris and Kenneth G. Smith. McGraw-Hill Book Co., New York City. Cloth; 187 pages, 6¼x9½ in.; 86 illustrations. Price, $1.50.

To teach the fundamental principles of mathematics to shopmen is the aim of the book. The first twelve chapters are devoted to simple arithmetic, such subjects as fractions, percentage, ratio and proportion, areas and volumes, square root, etc., being treated in very simple style, with plenty of examples to drill the reader in the work. The balance of the book contains chapters on levers, tackle blocks, inclined plane and screw, work, power and energy, horsepower of engines and belting, mechanics of fluids, heat and strength of materials. This part of the book is intended to develop an ability to use simple formulas and simultaneously impart a knowledge of the principles of machines. The problems and applications relate largely to the metal-working trades, but are of just as much interest and value to the engineer in the power plant. References to everyday applications of the subjects mentioned and illustrations of familiar machines and devices make it easy for the reader to grasp the points of a problem. Due to the simple and direct treatment the book should prove specially valuable in home study.

THE THEORY OF ENGINEERING DRAWING. By Alphonse A. Adler, B. S., M. E. D. Van Nostrand Co., New York. Cloth; 289 pages; illustrated; 6x9 in. Price $2.

As the author states in his preface, the subject matter of this book is, in large measure, the same as dealt with in treatises on descriptive geometry. The chief aim is to correct the weaknesses considered to exist in the manner of presenting the theory of descriptive geometry.

The title is very appropriate considering that descriptive geometry applies more to engineering than to any other field, and because the author treats of its relation to engineering. Contrary to the usual manner of working up the subject by starting in with definitions and considerations of the orthographic projections of a point and going thence into the theory of the projection of lines, surfaces and solids, the author reverses the process by first considering a concrete object and then working up to the theory. The purpose of this method of presentation is to make it easy to understand each successive step of the subject.

The book is divided into four parts Part I treating oblique projection and orthographic projection, which includes axonometric projection; Part 2, an interesting variety of problems relating to the line, the point and the plane; Part 3, problems in perspective, and Part 4, shading.

The book will be found very useful to those who have to do with or are interested in the theory and commercial application of drawing. The volume is well adapted to home study, as a number of questions are given after and relating to each chapter.

THE METRIC SYSTEM. By Henry G. Bayer. Ninety-four 4x6½-in. pages; 13 illustrations in the text; 26 tables of equivalents. Price, 85c. Henry G. Bayer, 232 Greenwich St., New York, N. Y.

This little book has been prepared to furnish in useful form to the student, engineer and manufacturer, needful data in regard to the metric system. The author is an enthusiastic advocate of the system, but this does not blind him to the strength of the arguments in opposition; in fact these major arguments are briefly and accurately stated in the opening paragraph on page 11.

The history of the system, its definitions, rules, etymology, 26 conversion tables and explanatory problems are given. The general heads are: Measures of Length, of Surface, of Volume, of Capacity, of Weight (mass), of Value; Miscellaneous Measures Used in Physics and Engineering, and Thermometers.

The tables, which are among the most important and perhaps the most useful features have been worked out with care and completeness. They alone are sufficient to recommend the book, for too often the engineer exasperatingly fails to find the particular equivalent of some compound English unit when he wishes to make a conversion to the metric system. The book will be found of value to anyone having occasion to use the metric system.

LEFAX. Loose leaf engineer's reference book founded by John Clinton Parker, published by the Standard Corporation, Pennsylvania Bldg., Philadelphia, Penn. A year's subscription, $1. Size of pages 3¾x6¾-in. Loose leaf binder, leather covered, 90 cents.

With the evident idea of giving the engineer ultimately all the data which he needs to keep on hand, in the shape of loose-leaf sheets, so that he need carry for reference only such pages as are most important to him, J. C. Parker has hit upon the scheme which has now borne fruit in "Lefax," (Leaf-facts).

It takes care not only of the data sheets published by the Standard Corporation, but of the engineer's own notes, and of data published in periodicals, for such may be filed on blank sheets of the same standard size as the data sheets, and indexed among them. Clippings can be so pasted or a synopsis of the article written on the blank sheet. Subscribers are invited to suggest subjects for data sheets, and such as furnish data sheets which are accepted will be paid for them at the rate of $5 a sheet, provided such information is original.

On each sheet a filing index is printed, which can be modified at the individual user's pleasure.

In the sample sheets sent out are tables of logarithms, circumferences and areas of circles, the 1912 standard of flange fittings, actual evaporation per boiler horsepower and factors of evaporation for saturated steam, properties of saturated steam from the new Marks & Davis tables, actual evaporation per boiler horsepower and factors of evaporation of superheated steam, 100 deg. superheat, sheet metal and wire standard gages, and an example of the advertising sheets sold to manufacturers of equipment for presenting data of interest to engineers. The one given among the samples is from the Parker Boiler company, and describes the design of a Parker downflow boiler.

PRACTICAL MATHEMATICS, PARTS I, II AND III. Bound separately. By Claude Irwin Palmer. Published by the McGraw-Hill Book Co., New York City. 5x7 inches. Cloth. Price 75 cents per volume.

These three books are the first of a series of four volumes treating of mathematics. The series is the outgrowth of the course in mathematics given the evening classes of the Armour Institute of Technology. These little books are well adapted to the needs of practical men who must get a working knowledge of mathematics by home study.

Part I (138 pages) begins with a review of arithmetical terms and work which nearly every man has learned in the lower grades of the grammar school.

The following chapters take up common and decimal fractions, weights and measures, percentage, ratio, proportion, powers and roots. The exercises consist of working out problems that are met with in every day work in the shop, engine room, and in the building trades.

Part II (150 pages) treats solely of the essentials of geometry. Plenty of exercises are given so the student may get a good training in the working of problems relating to triangles, circles, prisms, cylinders, spheres, pyramids, cones, frustrums, etc.

Some examples in geometry relating to the work of the carpenter, machinist, mason or engineer, are fully explained and worked out. This book concludes with a list of 61 geometrical formulas and five tables.

Part III (176 pages) takes up algebra. The author treats the subject so feared by most practical men in a manner which is necessarily lengthy on account of the simplicity with which the matter must be presented.

At the end of each lesson a number of examples are given with their answers so that the student may know whether or not he has worked his problems correctly by reference to these answers.

Engineers will find Chapter 15 of this volume to be of a special interest to them in that it takes up the essentials for plotting graphics.

THE ELEMENTS OF HEATING AND VENTILATION. By Arthur M. Greene, Jr. John Wiley & Sons, New York, 1913. Cloth; 324 pages, 6x9¼ in.; 223 illustrations, tables. Price $2.50.

The book is a general treatment of heating and ventilating systems, covering hot-air, hot-water and direct and indirect steam heating systems, particular attention being given to residence heating. Formulas for the amount of air required for ventilation are deduced and supplemented by tables from various authorities. This data is naturally followed by a description of humidifiers and washers for cleaning the air. The subject of heat losses from rooms is given considerable space. The various common forms of building construction are illustrated with data and formulas for calculating the heat loss and the heat required for ventilation. Radiators and valves of various types are illustrated and the subject of heat transmission through radiators is taken up at length. In direct steam heating, the single-pipe, double-pipe and over-head systems are compared and in the chapter following, gravity hot-water systems are given attention.

Indirect heating is limited to 65 pages. To illustrate the natural-draft system, the size of the flues and ducts and the amount of the heating surface is calculated for a two-story house. The plenum or forced-draft system is applied to a school house. The flow of air is taken up and formulas for the drop of pressure in pipes and bends are given. Methods for measuring the quantity of air by pitot tubes, standard orifices, anemometers, venturi meters and electric meters are explained and tables are given of pressures and the velocities of air and the diameters of pipes required for various capacities. Several forms of fans and housings are illustrated with problems showing the method to be pursued in determining the size of a fan for a given case.

In the section devoted to hot-air heating, a system is laid out and all calculations made for a two-story house. The details of furnaces and boilers and the points governing their selection are taken up and following is a description of the relative merits of steam, hot-water and hot-air systems.

District heating with steam and hot water is compared and the concluding chapter is devoted to thermostatic temperature control and the use of air for drying.

Due to the number of subjects discussed, the treatment is necessarily brief and, as the book deals mostly with house heating and ventilation, its value as a general reference book is limited in the same proportion to this particular branch of the art.

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Proposed Maine License Law

In the Maine Legislature an act to provide for licensing engineers of stationary engines has recently been proposed by Mr. Harman, of Stonington. The bill contains 10 sections.

Section 2 declares that whosoever desires to act as engineer of a stationary engine shall apply for a license therefor to the secretary of state. The application must show the applicant's total experience and be accompanied by the fee of $2.

The bill will allow an owner or user to operate a steam boiler for one week without a license. If the secretary of state decides that the applicant has properly answered all the questions in the application, and "the applicant has had the required two years' experience as a fireman," a license will be granted to operate a stationary engine in any part of the state of Maine for a period of two years unless revoked for incompetency or negligence. He must be "a citizen or naturalized citizen of the United States," of good moral character and of sober habits.

It is also proposed that a license may be renewed at the end of the two years without the $2 fee for renewal if the application is made within six months from the date of expiration of the license.

A violation of these rules subjects the engineer to a fine of not less than $100 or more than $500, imprisonment for not less than 30 days or more than three months, or both. The trial justice, the municipal and the police courts will have jurisdiction of offenses under the act.

Employees Share in Profits

Profit-sharing by employers with their employees received a practical endorsement at a banquet held in the Hotel Walton, Philadelphia, Monday evening, Feb. 17, at which were gathered the thirty-two associates and employees of the New York and Philadelphia offices of the architectural and engineering firm of Ballinger & Perrot.

Walter F. Ballinger presided, and gave a brief address explaining the profit-sharing plan which had been decided upon. This division will be upon a two-fold basis; the first, the division of a certain sum set aside from the profits in which all the employees will share equally, irrespective of their salaries. Second, there will be a division of an additional amount based upon the proportion which each employee's salary bears to the total amount of salaries paid.

Mr. Ballinger announced that the past year had been the most successful the firm had ever handled, and it was due, in a great measure, to the efficiency and coöperation of the employees. Emile G. Perrot then distributed checks to the employees, representing their proportion of the profits of the past year on the basis outlined.

Profit-sharing has proved so satisfactory that the firm has decided to extend the plan to all of its employees, from the highest to the lowest, believing thereby to obtain an increased coöperation and efficiency throughout their entire force. No pretense is made that the plan is a species of philanthropy, it being their firm belief that fair treatment of the employees is of mutual benefit to employer and employee.

Members of American Institute of Steam Boiler Inspectors Dine

The annual dinner of the New York section of the American Institute of Steam Boiler Inspectors was held at the Oak Hotel on Friday evening, Feb. 21, to commemorate the second anniversary of the founding of the society. About sixty members and guests were present and although not a large assemblage, they made up for lack of numbers in the enthusiasm shown.

As a mark of appreciation, the society presented its retiring president, J. W. Winter, with a handsome gold watch; T. T. Parker making the presentation.

Among the speakers were several representatives from other cities. These included Mr. King and Mr. Malloy, of Boston. The latter told of the Boston section of the American Institute of Steam Boiler Inspectors, its organization, educational program and policy.

Philadelphia was represented by Mr. Lukens, chief inspector of boilers for that city. He made a plea for uniform boiler laws and incidentally mentioned that since the Philadelphia laws had been in force there had not been a boiler explosion in that city.

The government service was represented by Mr. Arklebaum, who made a few remarks.

Mr. Fogarty, of Brooklyn, spoke from the viewpoint of the boiler manufacturer. He told how difficult it was to build a high-grade boiler to compete with an inferior one under existing conditions, and also threw some light on the present city inspection of steam boilers by the police department. He strongly advocated the adoption of state boiler rules for New York similar to those now in force in Massachusetts and also advocated giving insurance inspectors the power to shut down a boiler which they did not believe safe.

F. G. Shaw, the newly elected president, then spoke of the aims of the society and heartily endorsed what the other speakers had said regarding uniform boiler laws. Upon the motion of T. T. Parker, the society adopted a resolution providing for the presentation of a bill to the present legislature, for boiler laws based on the Massachusetts rules.

Thomas Wilmott, a member of the Assembly from Brooklyn, pledged his support of such a bill when drawn up.

Brief speeches were also made by a number of other members and guests and R. A. Thompson was reëlected secretary for the ensuing year. A splendid musical program added to the attractions of the evening.

MEMBERS AND GUESTS AT DINNER OF AMERICAN INSTITUTE OF STEAM BOILER INSPECTORS

SOCIETY NOTES

A meeting of the American Institute of Consulting Engineers to further discuss "Professional Relations" will be held at the Engineers' Club, 32 West Fortieth St., New York City, Tuesday evening, Mar. 11, at 8 p.m. The discussions of L. B. Stillwell and S. Whinery at the last meeting were printed in "Engineering News" of Jan. 23, page 155, and it is hoped that the members will read these articles and come prepared to speak. There will be an informal dinner at 6:30 at the same place just before the meeting.

In a paper read by K. Cox on "Internal Combustion Engines" recently, before the Yorkshire section of the Institution of Electrical Engineers, the author established comparisons between three different classes of plant as follows: In regard to capital cost, that of a 650-kw. turbine generating plant amounting to £8850, as against £960 for a gas engine installation and £11,080 for a Diesel engine installation of similar power. The running costs for the three plants under equal conditions were as follows: Steam plant, 0.232d. per brake hp.-hr., and 0.348d. per kw.-hr.; gas engine plant, 0.195d. and 0.292d. respectively, and Diesel engine plant, 0.268d. and 0.047d. respectively.

OBITUARY

PATRICK W. CANTWELL

Patrick W. Cantwell, a boiler maker, for many years in the department of water supply at the Ridgewood pumping station, Brooklyn, N. Y., and long the president of the Boiler Makers and Iron Ship Builders Association of America, died,

Feb. 7, at his home in Greenpoint. He was born in the Eastern District 54 years ago and lived in Greenpoint 30 years. He is survived by his widow and three daughters.

CARL GUSTAF PATRIK de LAVAL

Carl Gustaf Patrik de Laval, of Stockholm, Sweden, died recently in the sixty-seventh year. He was born at Bläsenborg, County of Orsa, Province of Dalarne, Sweden, May 9, 1845. His father was Capt. Jacob de Laval, of the Swedish Army; his mother's maiden name was Johanna Martin. A mechanical taste became evident in his childhood, for he loved to take apart and put together pieces of mechanism, such as watches and gun locks. His parents were wise enough to let his inclination develop. Having continued his home instruction until the age of 12, they sent him to school in Salun, Sweden, and at 18 he was a student in the University of Upsala, Sweden. It is said also that he was enrolled in the University of Stockholm. At all events, he reached his college graduation with high honor in 1866. De Laval's first employment was with the "Stora Kopparberg," a mining concern, where he found that he needed a better scientific education and thereupon returned to Upsala, taking up postgraduate study in chemistry physics, mathematics, etc. In 1872, having attained the doctorate of philosophy, he reentered the service of the "Stora" company. About this time he journeyed to the Harz Mountains, in Germany, to study the manufacture of sulphur or sulphuric acid, a plant for the production of which was projected at Salun.

Going to Germany in 1875, he became mechanical engineer for the Kloster Iron Works, where he invented a sort of sieve for improving the distribution of air in bessemer converters. He also invented a new apparatus for galvanizing purposes. More significant were his experiments with the centrifugal cream separator, which perhaps has made his name even more widely known than did his steam turbine later.

Prof. Lefeldt, it is fair to mention, was working with the separator idea at about the same time as de Laval. The latter, seeing its possibly enormous utility, and having failed to persuade the Kloster concern to take up its manufacture, surrendered his employment with them, in 1877, and went to Stockholm to exploit the invention himself. Since that time, the sales of his cream separator have amounted probably to a million machines. The mechanism has undergone important improvements over the original pattern, among them being a continuous delivery of cream and skimmed milk from their respective outlets. De Laval made other inventions in the dairy business, including the "Lactotrite," "Emulser," and "Continuous Churn." The "Lactotrite"—which we understand to be the same as the "Lactometer"—is an instrument for automatically analyzing milk to show its quality. Previously it had been customary to determine the proportion of cream, but de Laval's instrument accurately indicates its richness in butter fat.

Following the establishment of his cream-separator industry, Dr. de Laval founded at Olofström, Sweden, a factory for stamping out steel utensils, which has grown into a very important national industry. At heavy cost, he experimented with a steamboat to develop high speed on a new principle, but was not very successful, and, although confident that he had the right idea, he decided to drop it and put his time and money into the steam turbine.

As long ago as 1870, he had experimented with a nozzle for a steam sand blast to be used for making mine borings, the edges of which nozzle were so inclined as to cause rotation. This, though of little account in itself, turned his mind in a fruitful direction. About 1882, he constructed his first steam reaction turbine. It was a simple one, on the principle of Hero's engine, the rotor being essentially an S-shaped outward-flow pipe. It was geared with friction bevel wheels whose pressure was produced by the axial thrust of the turbine wheel. This turbine did not expand the steam enough to run at very high speed.

In one or more of these early experimental turbines he used water (condensed steam) for bearing and stuffing-box lubrication. The shaft rested on a stepbearing inclosed in a vessel filled with water under high pressure, which kept the shaft up to a certain level. If it rose higher, the water escaped, relieved the pressure and allowed the shaft to drop back. By this construction a considerable power was developed, but the bad lubrication caused heating at the higher speeds, and it seemed impossible to make a tight packing between the steam inlet and the fast rotating wheel.

To introduce the steam without impeding the rotation of the wheel, de Laval tried the plan of initially converting a part of the energy into kinetic form. This gave it the velocity required when it reached the rotor and the consequent lowering of pressure allowed a play of one-half to one millimeter without leaking.

Doctor de Laval's first commercially practical turbine was built in 1883, and was used to drive a cream separator, directly connected. Its efficiency was low, due to the slow peripheral speed and imperfect steam expansion.

It was not until 1888 that he arrived at the system of completely expanding the steam and employing it in an axial turbine. For this purpose, he used a multiplicity of nozzles in which the steam expanded to atmospheric pressure and produced a corresponding velocity. The wheel was designed for this velocity and the turbine was an axial impulse one which ran at a high number of revolutions per minute. At the close of 1888, de Laval invented the flexible shaft that allowed the wheel to revolve at a speed limited only by the strength of the material, thus perfecting the principle of his turbine. Tremendous were the practical difficulties that he surmounted in designing turbine wheels so constructed and balanced as to revolve around their centers of gravity at 16,000 to 30,000 r.p.m. without destructive side pressure on the bearings and to produce horsepower ranging up into the hundreds. In experiments he obtained even greater speeds.

De Laval formed his turbine business into a large incorporated company at Stockholm, Sweden, which had affiliated concerns in many countries, including France, Germany, England, Netherlands, Austria-Hungary, Russia and the United States. The number of his turbines that have been manufactured runs high into the thousands. They are employed for direct connection to dynamos, centrifugal pumps, blowers and other machines. Something like one in seven has been built for use on shipboard.

DOCTOR DE LAVAL

The demand by the turbine for high pressures of steam led de Laval to experiment with boiler designs, in which he applied new principles and aimed at absolute freedom from explosions. His boldness and unconventionality as an inventor may be judged from the fact that he worked with steam pressures of several thousand pounds per square inch.

He was an engineer of great versatility. One of his more recent inventions was a method for making iron pipe. He busied himself also with metallurgy, among his later patents being those on electric furnaces for direct production of metals from the ores, a method especially valuable in a country like Sweden, which is rich in water powers. A zinc-smelting plant on the de Laval principle was established at the Trolihätte waterfall, and still later he patented a furnace for smelting iron from a pulverized charge.

De Laval's personality was one which drew to him the respect and love both of his social peers and of his employees, and his valuable citizenship was recognized by the bestowal of national offices. He was elected a deputy to the Swedish Parliament, 1888-90, and later became a member of the Senate. De Laval's honors and decorations were many, he having received the Cross of Commander of Wasa, and the insignia of Knight of the First Class of the North Star. He was a member of the Swedish Royal Academy of Science, which presented him with a gold medal in 1892, and an honorary member of the Royal Academy of Agriculture. In 1904, the Association of German Engineers awarded him the Groshof medal for his pioneer work on the steam turbine.